Integrated organic Rankine cycle (ORC) and heat pump (HP) systems for domestic heating

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Abstract

Space and water heating represent a significant share of the overall energy consumption in the domestic sector. Decarbonising heat, though challenging, is acknowledged as having a key role to play (as exemplified by the Domestic Renewable Heat Incentive launched in 2014 in the UK, amongst other) in achieving emissions reduction targets and alleviating problems related to energy shortage and environmental deterioration. Novel, highly efficient heating technologies have attracted increasing interest in this context, in particular in regions with colder climates and higher heating demands. Specifically, thermally-driven heat-pumping technologies are a promising solution to meeting energy-efficiency targets by increasing the effective heat-to-fuel ratio (HFR) of heating systems. In this paper, thermally-driven integrated organic Rankine cycle (ORC) and heat pump (HP) systems are proposed for domestic heating applications, in which the ORC system is driven by heat from fuel (e.g., gas) combustion and generates power to drive an air-source vapour-compression HP system. A heat-transfer fluid is heated in the condensers of the two sub-systems to the required temperature for heat provision. Two system configurations with reversed heat-transfer fluid flow directions are presented and compared. Suitable, low global-warming-potential (GWP) working fluids for both the ORC and HP systems are considered and parametric optimisation is performed to determine optimal thermodynamic performance and system layouts. In a configuration in which the heat-transfer fluid flows first through the HP condenser and then through the ORC condenser in series, the HFR reaches values of 1.26-2.04 for air-source temperatures ranging from -15 to 15 °C and for heat provision temperatures from 35 °C to 60 °C. A performance enhancement up to 8-19% relative to the configuration with the heat-transfer fluid flowing in the reverse direction, i.e., through the ORC condenser and then the HP condenser in series, can be achieved. The specific investment costs of both configurations under typical conditions are around 600 £/kW_{th}, which indicates that the proposed systems are slightly higher but still economically competitive with existing HP products available on the market, thus demonstrating the potential of exploiting such novel systems for domestic heating in practical applications.

Keywords: domestic heating; ORC; heat pump; integrated system; thermo-economic

1. Introduction

Heat provision, including space heating and hot water, in the domestic sector represents a significant share of the overall energy consumption. It also accounts for a considerable fraction of harmful emissions. It is reported that domestic heating still continues to be dominated by fossil fuel-based heating units and less-efficient conventional electric heating technologies [1]. For example, in the UK about 90% of homes use gas boilers [2], which for a long time have been accepted as convenient heating source with a heat-to-fuel ratio (HFR) of around only 90% [3]. There has been an increasing effort from the government, academia and industry to decarbonise domestic heating to alleviate problems related to energy shortage and environmental deterioration, which is far more challenging compared to the power sector due to mostly decentralised generation. Among all the emerging technologies, thermally-driven heat-pumping systems have appeared as a promising solution given their high operating efficiency to replace the existing fossil fuel-based heating units by directly connecting to the burner/boiler.

Absorption, adsorption, and ejector heat pumps (HPs) as well as integrated systems coupling power cycles and HPs are typical thermally-driven heat-pumping options and extensive studies are available in this field. Xu et al. [4] presented a double-section absorption HP with LiBr-water as working pair for low-grade waste-heat recovery and a coefficient of performance (COP), the ratio of heat output to heat input, of 1.77 was achieved. Mirl et al. [5] designed an ammonia-water absorption HP system to meet targets such as high efficiency, high heating temperature and high system compactness. The maximum COP reached by the prototype was 1.42. Boman et al. [6] developed thermodynamic and heat transfer models for initial screening of working pairs used in adsorption HPs, indicating that AX-21+ammonia was the best option for heating and further research was recommended on MOF+alcohol working pairs. Seol et al. [7] presented models to predict performance of adsorption HPs with A-type silica gel and Wakkanai siliceous shale (WSS) as adsorbents, with results revealing that the COP could be improved by 6-17% when WSS was used relative to the system using A-type silica gel. Taslimi Taleghani et al.

[8] investigated a transcritical CO₂ ejector HP system and the optimisation of ejector geometry parameters could increase the COP and heating capacity by 17% and 20%, respectively. Liu and Lin [9] analysed the performance of a dual-temperature ejector-compression HP with a conventional system as the baseline case. The results showed that the ejector-based system yielded both better thermodynamic and economic performance.

Integrated systems that couple power cycles and HPs are also an attractive alternative and have been extensively investigated, in particular the concept of using an organic Rankine cycle (ORC) to drive an air-source vapour-compression HP system, as ORC has been proven as an effective heat-to-power conversion technology for low-to medium-temperature heat sources [10,11]. Liang et al. [12] proposed a gas-fuelled hot water system consisting of a premixed gas burner, an ORC and an air-source HP. The comprehensive numerical analysis demonstrated that an overall FHR up to 1.47 could be obtained. The authors [13] continued to develop dynamic models to determine optimal control strategies for two system configurations to achieve the maximum energy efficiency at fluctuating ambient conditions. Zhao et al. [14] investigated the performance of an ORC-HP combined system for diesel engine waste-heat recovery for combined heat and power (CHP) generation and a maximum total efficiency (defined as the gross heat and power output to the heat recovered) of 48% was achieved.

Compared to other thermally-driven HPs, integrated ORC and HP systems offer advantages including competitive thermodynamic performance as well as high operating flexibility under various conditions. Moreover, such systems can also be easily retrofitted for CHP provision. These CHP systems can cover flexible demand for heating and power, by adjusting the power input to the HP compressor, which further extends the application in the domestic sector. Previous studies mainly focused on the thermodynamic performance of integrated ORC and HP systems, while few have evaluated the thermo-economic performance over a wide range of operating conditions to indicate the potential to compete with other heat-pumping technologies. In this paper, we investigate such integrated systems for domestic heating, in which the ORC system is driven by heat from fossil fuel combustion and generates power to drive the HP system, and the heat-transfer fluid absorbs heat from the condensers of the two sub-systems. Two system configurations with opposite heat-transfer fluid flow directions are presented and compared, with parametric optimisation performed to determine the optimal thermodynamic performance at various operating conditions. The economic viability and benefits are also assessed to evaluate the potential of exploiting such systems in practical applications.

2. Methodology

2.1 System configurations

In the integrated ORC and HP systems, the ORC sub-system is driven by heat from fuel combustion and generates power to drive the HP compressor. The heat-transfer fluid (cold water) flows to the condensers of the two sub-systems in series to absorb heat. It should be noted that if the temperature of the heat-transfer fluid at the system outlet is lower than the required temperature for heat provision, some additional thermal energy from the fuel combustion is added to the heat-transfer fluid in order to reach the required temperature. Otherwise, a cold flow of water can be introduced to mix with the main stream of the heat-transfer fluid to cool it down to the set point temperature. Two configurations of the integrated ORC and HP system with reversed heat-transfer fluid flow directions are presented, of which the schematic diagrams are shown as below. Configuration 1 (see Figure 1(a)) represents the system with heat-transfer fluid flowing to the ORC condenser first and then flowing to the HP condenser, while Configuration 2 (see Figure 1(b)) shows the system with heat-transfer fluid flowing to the HP condenser first and then to the ORC condenser. The two-system configuration have been presented in Ref. [13] and control strategies have been studied, while in this paper we focus on the design analysis and thermodynamic and economic assessment over a wide range of operating conditions.

(a)

Figure 1. Integrated ORC and HP systems for domestic heating: (a) Configuration 1: heat-transfer fluid flows through the ORC condenser and then through the HP condenser, and (b) Configuration 2: heat-transfer fluid flows through the HP condenser and then through the ORC condenser.

2.2 Thermodynamic models

The ORC and HP thermodynamic models based on the first law of thermodynamics have been described in the authors' previous work [15,16]. The heat-to-fuel ratio is used to evaluate the thermodynamic performance of the integrated systems, which is defined as:

$$\text{HFR} = \frac{Q_{\text{HS}}}{Q_{\text{fuel}}} = \frac{\dot{m}_{\text{HS}} \cdot c_{p, \text{HS}} \cdot (T_{\text{HS, out}} - T_{\text{HS, in}})}{Q_{\text{fuel}}} \quad (1)$$

where Q_{HS} is the heat absorbed by the heat-transfer fluid (heat sink), Q_{fuel} is the heat provided by the fuel, $c_{p, \text{HS}}$ is the specific heat capacity of the heat-transfer fluid, and $T_{\text{HS, in}}$ and $T_{\text{HS, out}}$ are the temperatures of the heat-transfer fluid at the inlet of the first condenser and the outlet of the second condenser, respectively.

2.3 Heat exchanger models

Heat exchanger models are developed to determine the required heat exchanger areas that correspond to the optimised cycle and source/sink outlet conditions as well as to calculate the costs. The condenser of the two sub-systems and the evaporator of the ORC sub-system are modelled as brazed-plate heat exchangers, while the evaporator of the HP sub-system is modelled as a plate-fin heat exchanger, since it extracts heat from the ambient air. The heat exchanger area, A , is given by:

$$A = \frac{Q}{U \cdot \Delta T_{\text{lm}}} \quad (2)$$

where Q is the heat load, U is the overall heat transfer coefficient and ΔT_{lm} is the log-mean temperature difference.

For the single-phase heat transfer in the desuperheating and subcooling sections of the condenser and the preheating and superheating sections of the evaporator, the Nusselt number, Nu , is expressed as a function of the Reynolds number Re and Prandtl number Pr :

$$Nu = a \cdot Re^b \cdot Pr^c \quad (3)$$

with coefficients a , b and c . To calculate the coefficients, the model switches between correlations of Bogaert et al. [17] for Re lower than 1000 and Chisholm et al. [18] for higher Re . The same correlation is also used for the heat-transfer fluid stream in both sub-systems and the heat-source stream of the ORC sub-system.

For the single-phase heat transfer on the air side of the evaporator, the heat exchanger coefficient is obtained from the Colburn factor j , which is expressed as a function of the Reynolds number:

$$j = a \cdot Re^b \quad (4)$$

with the coefficients a and b obtained from Chang et al. [19] for airflow over louver fins.

In the condenser two-phase flow region, the single-phase Bogaert [17] and Chisholm [18] correlations are used with an equivalent Reynolds number as recommended by Thonon [20].

In the evaporator two-phase flow region, the heat transfer coefficient is obtained from:

$$h = \left(h_{cb}^2 + h_{nb}^2 \right)^{\frac{1}{2}}, \quad (5)$$

where h_{cb} corresponds to the heat transfer coefficient attributed to convective boiling and is obtained by switching between the Bogaert and Chisholm correlations, while h_{nb} corresponds to the heat transfer coefficient attributed to nucleate boiling and is obtained from Cooper's correlation [21].

2.4 Economic models

Component cost correlations acquired by Quoilin et al. [22] and Lecompte et al. [23] are used in this study, based on the Belgian market in 2010. The chemical engineering plant cost index (CEPCI) is used to update capital cost required to erect a power-cycle system from a past date to a later time. Based on the results of the analysis of the Department of Energy & Climate Change on HP costs [24], the factors accounting for costs are obtained and summarised. All adjusted correlations are presented in Table 1.

Table 1. Component cost correlations [16,22-24].

Component	Capacity	Cost (£)
Brazed-plate HEX	Heat exchanger area (m ²)	337 + 214 · A
Plate-fin HEX	Heat exchanger area (m ²)	270 · A
Compressor	Volume flow rate (m ³ /s)	144000 · V + 216
Expander	Volume flow rate (m ³ /s)	144000 · V + 216
Pump	Power (W)	512 · (P/1000) ^{0.46}
Air fan	Frontal area (m ²)	578 · A
Expansion valve	/	123
Miscellaneous hardware	/	20%
Assembly (e.g., welding)	/	20%
Profit margin	/	20%
Tax	/	20%

2.5 Conditions and assumptions

Thermodynamic and economic models of the integrated ORC and HP systems are developed in MATLAB and working fluid properties are acquired from NIST REFPROP [25]. An interior-point optimisation algorithm is used to maximise the HFR by optimising the cycle parameters, namely: (i) the evaporation/condensation temperatures; (ii) superheating/subcooling degrees; (iii) the working fluids mass flow rates in both sub-systems; and (iv) the air-source mass flow rate. The main operating conditions and assumptions are given below:

- (1) The heating capacity of the proposed integrated ORC and HP systems is 10 kW, which is an average peak heating demand of a UK household; the UK this is taken here as representative of a temperate climate with modest temperature changes, mild winters and moderate summers [26].
- (2) The heat-transfer fluid inlet temperature is set to be 10 °C as a typical value for cold water, and the outlet temperature varies from 35 °C to 60 °C, which covers the temperature range for domestic heating and hot water provision.
- (3) Therminol VP-1 is selected as the heat-transfer fluid that absorbs heat from fuel combustion and transfers it to the ORC system as a heat source, with inlet and outlet temperatures of 200 °C and 100 °C, respectively.
- (4) The energy conversion efficiency from fuel combustion to heat provided to the ORC is assumed to be 0.9, which is a common value for existing boilers. This is taken into consideration for the HFR calculation.
- (5) R245fa is selected as the ORC working fluid and R32 is selected for the HP.
- (6) Both the ORC and HP sub-systems are operated in subcritical region.
- (7) Isentropic efficiencies of the pump, expander and compressor are assumed to be 0.8.
- (8) The pinch point temperature difference of all heat exchangers is set to be 5 °C.
- (9) Heat losses of the system to the ambient are neglected and all processes take place at steady state.

3. Results and discussion

3.1 Performance under typical operating conditions

The operating conditions of the integrated ORC and HP systems are selected to be: (i) air source temperatures of 7 °C (the industrial standard for quoting the efficiency of an HP [27]) and -10 °C (typical low-temperature conditions); and (ii) heat provision temperature (heat-transfer fluid outlet temperature) of 35 °C (typical temperature for only underfloor space heating) and 60 °C (typical temperature required to supply both space heating using radiators and hot water) [28]. Therefore, four typical operating conditions are considered, i.e., (i) $T_{air} = 7$ °C, $T_{HS,out} = 35$ °C; (ii) $T_{air} = 7$ °C, $T_{HS,out} = 60$ °C; (iii) $T_{air} = -10$ °C, $T_{HS,out} = 35$ °C; and (iv) $T_{air} = -10$ °C, $T_{HS,out} = 60$ °C.

Figure 2 shows the thermodynamic performance of the two system configurations under the aforementioned typical operating conditions. Configuration 2, in which the heat-transfer fluid flows to the HP condenser first and then to the ORC condenser, provides a higher heat-to-fuel ratio under all conditions, in particular when a high heating provision temperature is required. Moreover, a higher air-source temperature and a lower heat provision temperature are favourable for both systems to yield better thermal performance. Under the first operating condition of $T_{\text{air}} = 7^\circ\text{C}$ and $T_{\text{HS,out}} = 35^\circ\text{C}$, the heat-to-fuel ratios of Configurations 1 and 2 are 1.60 and 1.77, respectively. Figure 2(b) shows a plot of the heat-transfer fluid (heat sink) intermediate temperature; this is the temperature of the water once it is out of the first condenser, before entering the second. Configuration 1, in which the heat-transfer fluid flows to the ORC condenser and HP condenser in series, provides a higher intermediate temperature, except under the first condition with the standard air source temperature (7°C) and low heat provision temperature only for space heating (35°C). Figures 2(c) and 2(d) show the thermodynamic performance of the two sub-systems. Higher ORC efficiency is achieved in Configuration 1 due to the low condensation temperature, since the heat sink with a low initial temperature (10°C) is used to cool down the ORC working fluid. On the other hand, a higher COP (defined as the ratio of heat output to the power input) of the HP subsystem is obtained in Configuration 2, due to a small temperature difference between the heat sink and cold source. The thermal loads of the two condensers are plotted in Figures 2(e) and 2(f), which indicate that the ORC condenser contributes more heat to the heat-transfer fluid in Configuration 1, in particular under conditions with a high heat provision temperature, while the HP condenser plays a more significant role in Configuration 2.

(a)

(b)

(c)

(d)

Figure 2. Thermodynamic performance of integrated ORC and HP systems for domestic heating under typical operating conditions: (a) heat-to-fuel ratio, (b) heat-transfer fluid (heat sink) intermediate temperature between the two condensers, (c) efficiency of ORC sub-system, (d) COP of HP sub-system, (e) thermal load of the ORC condenser, and (f) thermal load of the HP condenser.

Design parameters of both configurations under the four typical conditions are listed in Table 2. It is observed that in Configuration 2, the temperature difference between the HP condensation and evaporation temperatures is smaller than that of Configuration 1, which enables the HP in Configuration 2 to provide a higher COP.

Table 2. Design parameters of integrated ORC and HP systems under typical operating conditions.

		Configuration 1 - ORC-HP			
		$T_{air} = 7^\circ\text{C}$ $T_{HS,out} = 35^\circ\text{C}$	$T_{air} = 7^\circ\text{C}$ $T_{HS,out} = 60^\circ\text{C}$	$T_{air} = -10^\circ\text{C}$ $T_{HS,out} = 35^\circ\text{C}$	$T_{air} = -10^\circ\text{C}$ $T_{HS,out} = 60^\circ\text{C}$
ORC	Condensation temperature ($^\circ\text{C}$)	23	35	25	38
	Evaporation temperature ($^\circ\text{C}$)	150	150	150	150
	Superheating degree ($^\circ\text{C}$)	45	45	45	45
	Mass flow rate (kg/s)	0.017	0.022	0.020	0.025
HP	Evaporation temperature ($^\circ\text{C}$)	-2	-1	-18	-17
	Superheating degree ($^\circ\text{C}$)	0	0	0	0
	Condensation temperature ($^\circ\text{C}$)	38	59	37	59
	Subcooling degree ($^\circ\text{C}$)	11	14	8	10
		0.017	0.013	0.013	0.010
		Configuration 2 - HP-ORC			
		$T_{air} = 7^\circ\text{C}$ $T_{HS,out} = 35^\circ\text{C}$	$T_{air} = 7^\circ\text{C}$ $T_{HS,out} = 60^\circ\text{C}$	$T_{air} = -10^\circ\text{C}$ $T_{HS,out} = 35^\circ\text{C}$	$T_{air} = -10^\circ\text{C}$ $T_{HS,out} = 60^\circ\text{C}$
ORC	Condensation temperature ($^\circ\text{C}$)	37	57	36	56
	Evaporation temperature ($^\circ\text{C}$)	150	150	150	150
	Superheating degree ($^\circ\text{C}$)	45	35	45	37
	Mass flow rate (kg/s)	0.016	0.022	0.020	0.025
HP	Evaporation temperature ($^\circ\text{C}$)	-2	-1	-18	-18
	Superheating degree ($^\circ\text{C}$)	0	0	0	0
	Condensation temperature ($^\circ\text{C}$)	27	35	25	31
	Subcooling degree ($^\circ\text{C}$)	12	20	10	16
		0.017	0.014	0.013	0.011

3.2 Economic performance evaluation

The economic performance of the two system configurations is evaluated using the specific investment cost (SIC) as an indicator. It can be seen from Figure 3 at the first condition with air source temperature of 7°C and heat provision temperature of 35°C , the SIC of Configuration 1 is higher than that of Configuration 2, while Configuration 1 is economically preferable under all other conditions. The SIC of both configurations under all conditions are around 600 £/kW_{th} and the differences are not significant. A detailed market research was

conducted in the authors' previous work [29] to obtain the costs of available domestic electric HPs in the UK. The market research shows that a typical 10 kW_{th} air-source electric HP has an investment cost per unit of power of about 400-450 £/kW_{th}, which is obtained by averaging the cost of a number of different-performance heat pumps from a range of different manufacturers. In the integrated systems, the electric HP sub-system only covers a fraction of the 10-kW demand, therefore the components are smaller than those of a typical 10 kW_{th} electric HP. Although, as expected, the integrated ORC-HP system is slightly more expensive, it shows to be economically competitive with the existing heat-pumping products available on the market. The fact that this system is operated on heat from gas (which is much cheaper than electricity) demonstrates the potential for further investigation in practical applications. It should be also mentioned that the ORC-HP system could be run on hydrogen in the future, in line with net-zero emission objectives.

Figure 3. Specific investment cost of integrated ORC and HP systems under various typical operating conditions.

A detailed economic assessment of the two system configurations under the representative condition of $T_{air} = 7\text{ °C}$ and $T_{HS,out} = 60\text{ °C}$ is performed and the costs breakdown of all the components is shown in Figure 4. It can be seen that the dominant components in terms of costs for both sub-systems are the heat exchangers, in particular the two condensers. More precisely, the costs of heat exchangers account for ~60% of the total costs for both configurations. Moreover, the heat exchanger costs in Configuration 2 are slightly higher than those in Configuration 1, which results in a higher S/C as shown in Figure 3. It should be stated that the heat sink inlet temperature is low (10 °C) in this study and the proposed integrated systems are used to heat it up to the set temperature directly, therefore, large heat exchanger area is required to allow a large amount of heat exchange, which is therefore associated with a high cost.

Figure 4. Cost breakdown of all the components of the integrated ORC and HP system under the representative operating conditions of $T_{air} = 7\text{ °C}$ and $T_{HS,out} = 60\text{ °C}$.

3.3 Performance over wide range operation

This section focuses on the thermodynamic performance as well as the comparison of the two system configurations at the design point over a wider range of operating conditions. The air-source temperature is

varied from $-15\text{ }^{\circ}\text{C}$ to $15\text{ }^{\circ}\text{C}$ and the heat provision temperature is varied from $35\text{ }^{\circ}\text{C}$ to $60\text{ }^{\circ}\text{C}$ to cover the temperature range for space heating and domestic hot water. In Figure 5, the HFR of the two system configurations are plotted. For Configuration 1, the heat-to-fuel ratio ranges from 1.12 to 1.82, while it ranges from 1.26 to 2.04 for Configuration 2. Configuration 2 provides better thermal performance and the performance enhancement relative to Configuration 1 ranges from 8% to 19% over all the selected operating conditions. It is also clearly observed that higher air source temperature and lower heat provision temperature enable both systems to achieve higher heat-to-fuel ratios.

Figure 5. Heat-to-fuel ratio of integrated ORC and HP systems for domestic heating under a wide range of operating conditions: (a) Configuration 1: heat-transfer fluid flows to ORC condenser and HP condenser in series; and (b) Configuration 2: heat-transfer fluid flows to HP condenser and ORC condenser in series.

The comparison of ORC efficiency and HP COP of the two system configurations is shown in Figure 6. The ORC efficiency of Configuration 1 ranges from 15% to 17%, which is higher than that of Configuration 2 ranging from 12% to 15%. The HP COP of Configuration 2 ranges from 4.17 to 9.52, which is significantly higher than that of Configuration 1 ranging from 2.64 to 7.01, enabling Configuration 2 to yield a better overall thermal performance and a higher heat-to-fuel ratio.

Figure 6. ORC efficiency and HP COP of integrated ORC and HP systems for domestic heating under a wide range of operating conditions: (a) Configuration 1: heat-transfer fluid flows to ORC condenser and HP condenser in series; and (b) Configuration 2: heat-transfer fluid flows to HP condenser and ORC condenser in series.

4. Conclusions

Thermally-driven integrated ORC and HP systems for domestic heating applications have been investigated in this paper in which an ORC system is driven by heat from fuel combustion and generates power to drive an air-source vapour-compression HP system. In these systems, a heat-transfer fluid absorbs heat in the condensers of the two sub-systems for heat provision. Two system configurations with opposite heat-transfer fluid flow directions have been presented and compared in terms of both thermodynamic and economic performance under typical operating conditions with air source temperatures of 7 °C (standard temperature for quoting HP performance) and -10 °C (low-temperature conditions), and heat provision temperatures of 35 °C (for space heating only) and 60 °C (for both space heating and hot water). The results indicate that a higher ORC efficiency can be achieved by Configuration 1 (heat-transfer fluid flows to the ORC condenser and HP condenser in series) due to the lower condensation temperature, while a higher COP can be attained by the HP subsystem in Configuration 2 (heat-transfer fluid flows to the HP condenser and ORC condenser in series). Configuration 2 has been shown to provide a higher heat-to-fuel ratio under all conditions examined in this work, in particular at high heat provision temperatures. The specific investment costs of both system configurations under the 4 typical conditions are around 600 £/kW_{th}, which indicates that the proposed systems are slightly more expensive but still economically competitive with existing heat-pumping products available on the market. A wider range of operating conditions has also been considered, for which it was demonstrated that the heat-to-fuel ratio of Configuration 2 can reach values between 1.26 and 2.04, and a performance enhancement up to 8-19% relative to Configuration 1. The thermodynamic and economic assessments of the integrated ORC and HP systems demonstrate significant potential of exploiting such systems for domestic heating in practical applications.

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Nomenclature

Symbols:

a, b, c	coefficients for heat transfer calculation
A	area, m ²
c_p	heat capacity, J/(kg·K)
h	heat transfer coefficient, W/(m ² ·K)
j	Colburn factor
Nu	Nusselt number

P	power (W)
Pr	Prandtl number
Q	heat load, W
Re	Reynolds number
T	temperature, K
U	overall heat transfer coefficient, W/(m ² ·K)

Subscripts

cb	convective boiling	lm	log-mean temperature difference
HS	heat sink	nb	nucleate boiling
in	inlet	out	outlet

Abbreviations

CEPCI	chemical engineering plant cost index	HP	heat pump
CHP	combined heat and power	ORC	organic Rankine cycle
COP	coefficient of performance	S/C	specific investment cost
HFR	heat-to-fuel ratio		

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