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Integration of Low-Grade Heat from Exhaust Gases into Energy System of the Enterprise

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Abstract

In the paper is presented the way of Process Integration application for waste heat utilisation from exhaust gases streams with partial condensation. It is based on the construction of Hot Composite Curve representing the gaseous mixture cooling with accounting for the gas-liquid equilibrium of condensable vapour part. With Cold Composite Curve for streams requiring heating, the Pinch Point is determined. Then the structure of Heat Exchanger Network (HEN) for utilised Heat Integration into the energy system of the factory is developed accounting for the possible splitting of two-phase flow on gas and liquid streams and selection of plate heat exchanger (PHE) types for specific positions in HEN. The method is illustrated by a case study of heat utilisation from exhaust gases after superheated steam tobacco drying and flue gases from natural gas-fired boiler. The heat transfer areas of PHEs in HEN are optimised with the total annualised cost as an objective function. The payback period of the received solution is less than four months with a substantial saving of energy, reduction of greenhouse gases and other harmful emissions of combustion processes.

Keywords Exhaust Gas, Waste Heat, Process Integration, Plate Heat Exchanger

Declarations

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Conflict of interest

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Not applicable

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Not applicable

1. Introduction

The sustainable development of mankind is characterised by a growing need for energy. But even with widening use of renewable sources the primary way of energy generation is the combustion of fossil fuels with the emission of CO₂ and other harmful substances polluting the environment. In industry, the considerable part of this energy is wasted with outgoing streams, cooling utilities and losses to the environment. According to the analysis presented in paper by Papapetrou et al. (2018), the potential of waste heat available in EU only is up to 300 TWh/y, with 33% of low-grade heat at temperatures below 200 °C and 25% with temperatures 200-500 °C and the rest at below 1,000 °C. Most of this waste heat is available in industrial zones (Huang et al. 2017). The use of this wasted energy can limit demand for fossil fuel combustion, leading to conservation of resources, reduction of carbon dioxide emissions and pollution of the environment. There is a potential of global emissions reduction by 44% in 2035 with most share achieved with waste heat usage that results in savings of primary energy (Oluleye et al. 2016).

A considerable part of the waste heat is going to the atmosphere with different off- and flue gases in many industrial sectors. The temperature of exhaust gases and their composition is determined by the conditions in the process of their origin. In many cases, the exhaust gas or some of the exhaust gaseous mixture components can condense to the liquid state being cooled down to specific temperatures (Varbanov et al. 2005). In this process, along with sensible heat of gas cooling the latent heat of phase transition is released. The relative share of sensible and latent heats is determined by exhaust gas composition and final temperature of the cooling process. In the majority of practical applications, the condensable part of gaseous exhaust mixture is water vapour. Its latent heat is about 500 fold higher than heat capacity, and the heat of condensation can be a substantial part of total heat balance even at its small content in the mixture with not condensing gases. The typical examples of such exhausts are flue gases after different fuels combustion in furnaces, ovens, stoves, boilers etc. When the fuel contains sulphur, the condensing water with sulphur oxide in the gas can form sulphuric acid, that promoting corrosion of heat exchanger. To avoid this sometimes it is recommended to cool gas not lower than steam condensing temperature, as, e.g. 90 °C (Xu et al. 2013) for coal-fired boiler and for a new design of heat recovery from flue gas of such boiler (Jin et al. 2019). But in this case, a considerable amount of latent heat and water would be lost and with sulphur oxide discharged to atmosphere elevating its pollution. To avoid this and to recuperate latent heat with flue gas cleaning from harmful oxides and generate some amount of water the direct contact economisers were proposed, as, e.g. in paper by Zhelev and Semkov (2004) for boiler fired with coal of low sulphur content or direct contact condenser of special design (Priedniece et al. 2019), as also described in the paper by Cui et al. (2020), and for simultaneous reduction of NO_x emissions

from gas boilers in paper by Zhang et al. (2020). However, the advantages of recuperative heat exchangers make their use economically feasible even with heat transfer surface made from expensive materials, as is shown in paper of Terhan and Comakli (2016) for condensing shell-and-tube flue gas heat exchanger made of AISI316 quality stainless steel. The compact heat exchangers for this purpose are even more promising (Shi et al. 2011); they are smaller in volume and require much less material for heat transfer surface as it discussed in the book by Klemeš et al. (2015). The use of corrosion-resistant material allowed the use of latent heat in the evaporator of heat pump for heating water of District Heating system (Vannoni et al. 2021) or also to elevate temperature in heat pump described in the paper of Bamigbetan et al. (2019). The use of recuperative heat exchanger allows the generation of electricity with Organic Rankine Cycle (ORC) and create cryogenic power generation system using sensible and latent heats of flue gas and liquefied natural gas (LNG) cold energy (Yu et al. 2021). In all cases of steam condensers applications, their design must be made for condensation in the presence of non-condensable gas (Li et al. 2019). The potential of waste heat utilisation from flue gases much depends on the steam content in the exhaust gas. It depends on fuel type and method of its combustion. By the data of different publications, it can vary from 5.5 % on volume for a coal-fired power plant in Poland (Więclaw-Solny et al. 2014) to 25.5 % for Australian brown coal-fired power plant (Dave et al. 2011) and 30 % for industrial waste incinerator (Aouini et al. 2014). The content of sulphur, nitrogen and carbon oxides is also different and can require specific material for the heat transfer surface of the condenser. The decision to use waste heat recovery method in specific conditions must be made on detailed technical and economic analysis of different available options (Ma et al. 2016).

Beside flue gases at power generation, there is a lot of heat wasted with exhaust gases in chemical, petrochemical, food, transport, communal and other sectors. It is exhaust gases from buildings space heating (Dudkiewicz and Szałański 2020), commercial kitchens (Wang et al. 2020b), transport heavy vehicles (Daccord 2017), light-duty vehicles (Wu et al. 2020), ceramics treatment in kiln (Montorsi et al. 2018), drying of food products with air (Julklang and Golman 2015), spray drying with air (Patel and Bade 2020), steam drying of algae (Aziz et al. 2013), spray drying for particles production (Julklang and Golman 2015). In all these applications the content of vapour in the exhaust gas can vary significantly and so the share of sensible and latent heat for recovery. The drying technology where the latent heat in exhaust gases is predominant over sensible heat is superheated steam drying (Li et al. 2016). It is frequently used for drying of different materials: food, wood, paper, coal and sludge. One example of heat utilisation after superheated steam drying is described in the paper by Gariev et al. (2014).

The efficient utilisation of heat from exhaust gases requires finding the best way of its Integration into the existing energy system. The Process Integration (PI) (Klemeš 2013) is an efficient tool for this purpose. It has been successfully extended into a number of implementations as, e.g. property-based resources conservation networks (Saw et al. 2011), bioenergy supply chain (Foo et al. 2013), for hybrid power (electricity) systems (Wan Alwi et al. 2012) and Integration of energy, water and environmental systems (Baleta et al. 2019). Implementation in the Heat Integration for utilisation of exhaust heat from flue gases in contact condensers of this methodology was used in paper (Priedniece et al. 2019) and for spray drier with exhaust gas heat recuperation by incoming air with the use of Pinch Analysis (Patel and Bade 2020). It has shown the usefulness of Process Integration methodology for finding the best solutions in heat recovery; however, it still needs an extension the features of this process in surface condensers. The Process Integration solutions can be much improved by using compact heat exchangers with intensified heat transfer (Klemeš et al. 2015). One of the efficient types of such exchangers is a plate heat exchanger (PHE). Due to considerable heat transfer enhancement in channels of complex geometry PHE have much higher overall heat transfer coefficient than traditional shell and tube heat exchangers and requires much less metal for the production of the heat transfer surface. It allows obtaining economically viable solutions with its fabrication from costly corrosion-resistant materials, such as different grades of stainless steels and other alloys. It allows the use of PHEs in applications of Process Integration in severe conditions of chemical industry (Tovazhnyansky et al. 2010), including acids production (Kapustenko et al. 2009), post-combustion capture of carbon dioxide (Perevertaylenko et al. 2015) and other cases. PHEs have low fouling tendencies (Crittenden et al. 2015). Some PHE types can be used as surface condensers of pure vapour (Arsenyeva et al. 2011), of multicomponent vapours mixtures (Tovazhnyansky et al. 2004) and of vapour from its mixture with non-condensing gas (Kapustenko et al. 2020).

This paper is implementing the Process Integration method (Klemeš et al. 2018) to find an efficient solution for utilisation of waste sensible and latent heat from exhaust gases with the use of recuperative heat exchangers. It accounts for the nonlinear character of enthalpy and heat capacity flowrate dependence from the temperature in gaseous mixture streams with partial condensation of one of its component. The method is illustrated with a case study where utilisation of heat from boiler flue gases and heat of gas exhausting from the superheated steam drying process is considered. The utilisation of heat is performed with the use of PHEs of specific types appropriate to each position of their application in the heat exchanger network.

2. Methodology

The stream of the gaseous mixture at a certain temperature contains the heat of its components, which includes sensible heat and latent heat of these components phase transition to a liquid state. With the cooling down of gaseous mixture to lower temperature, the heat contained in stream diminishes. If there is no phase change of the components, only sensible heat is reduced in almost linear dependence of temperature. The proportionality coefficient is equal to the sensible heat capacity of gaseous mixture that usually has a small dependence on temperature. On Composite Curves in PI, it is depicted by inclined segments of straight lines. In case of phase change of pure saturated vapour at constant pressure, internal heat of its stream not depends on temperature and is depicted as horizontal segments of straight lines on Composite Curves. However, when the gaseous mixture stream contains vapour component which saturation temperature is equal to the temperature of stream, the latent heat contained in it changes not directly proportional to temperature. This change is following to equilibrium between condensing vapour and its condensed liquid. The relation is not linear, and the process on Hot Composite Curve is depicted by a segment of the curve. This feature of condensing gaseous streams must be accounted in Pinch Analysis, as it can influence Pinch position and minimal temperature difference there.

The heat released with gaseous mixture stream cooling in heat exchangers must be transferred to cold streams that require heating. For estimation of all heat that can be taken from hot streams at different temperatures in Process Integration, the Hot Composite Curve can be used. This curve for exhaust gases must account for all forms of heat generated in the process of cooling: sensible heats of cooling not condensed gas and vapour; the heat of already condensed liquid; latent heat of the vapour condensed in considered temperature interval. The decrease of enthalpy in the gaseous mixture stream during its cooling with small temperature drop ΔT is determined as follows:

$$\Delta H = (G_g \cdot c_{p-g} + G_v \cdot c_{p-v}) \cdot \Delta T_{mx} + \Delta G_v \cdot r_v + G_{cn} \cdot c_{p-cn} \cdot \Delta T_{cn} \quad (1)$$

were G_g , G_v and G_{cn} are mass flowrates of non-condensing gas, vapour and liquid condensate, kg/s; ΔT_{mx} and ΔT_{cn} are the temperature changes of gaseous mixture and condensate, K; c_{p-g} , c_{p-v} and c_{p-cn} are specific heat capacities of non-condensing gas, vapour and condensate, J/(kgK); r_v is the latent heat of vapour condensation, J/kg.

With the use of Eq.(1) the Hot Composite Curve can be build assuming that: (1) for stream temperatures high than the saturation temperature of its vapour component no phase transition happens and only sensible heat must be accounted; (2) after the gaseous mixture stream is cooled

down to saturation temperature of its vapour component it is keeping to be saturated at all lower temperatures; (3) the overcooling of condensate is not considered, and Eq.(2) holds true.

$$\Delta T_{cn} = \Delta T_{mx} \quad (2)$$

The overcooling of condensate is accounted on heat exchangers design stage, as it is dependant of heat exchanger construction features and flow structure in it.

The vapour in the gaseous mixture has its volumetric fraction ε_v and partial pressure P_v , according to which the saturation temperature is determined by equilibrium conditions in K (the saturation temperature in centigrade is $t_s = T_s - 273.15$ °C):

$$T_s = T_s(P_v) \quad (3)$$

The vapour volume fraction is directly linked to its mass fraction and mass flowrates of vapour and non-condensing part of the mixture. The non-condensing gases in the mixture can be regarded as ideal gases, and for them, the Equation of State for ideal gas has the following form:

$$P_g \cdot V_{\Sigma} = R_g \cdot T_{mx} \cdot G_g \quad (4)$$

here V_{Σ} is the volumetric flow rate of the gaseous mixture, m^3/s ; P_g is the total, partial pressure of non-condensing gases in the mixture, Pa; R_g is a specific gas constant of non-condensing gases as their mixture, J/(kg K), determined as follows:

$$R_i = \frac{R}{\mu_i} \quad (5)$$

where $R=8314.5$ J/(kmol K) is the universal gas constant; μ_i is molar mass of the individual gas or in our case non-condensing gases mixture, kg/kmol.

The vapour at saturation is considered as a real gas with the application of real gas Equation of State Eq.(6). It is expressed in the same form as Eq.(4) with the introduction of compressibility factor z_v and specific gas constant R_v as follows:

$$P_v \cdot V_{\Sigma} = R_v \cdot z_v \cdot T_s \cdot G_v \quad (6)$$

From Eq. (6) divided on Eq. (4) follows:

$$\frac{P_v}{P_g} = \frac{R_v \cdot z_v}{R_g} \cdot \frac{G_v}{G_g} \quad (7)$$

For calculation of the compressibility factor z_v the real gas Equation of State proposed by Peng and Robinson (1976) is used:

$$P_v = \frac{R \cdot T_s}{v_v - b} - \frac{a(T_s)}{v_v \cdot (v_v + b) + b \cdot (v_v - b)} \quad (8)$$

Where

$$a(T_s) = 0.45724 \cdot \frac{R^2 \cdot T_c^2}{P_c} \cdot \left\{ 1 + k \left[1 - \left(\frac{T_s}{T_c} \right)^{0.5} \right] \right\}^2; b = 0.0778 \cdot \frac{R \cdot T_c}{P_c};$$

$$k = 0.37464 + 1.5422 \cdot \omega + 0.26992 \cdot \omega^2$$
(9)

Here v_v is vapour molar volume, $m^3/kmol$; T_c is its critical temperature, K; P_c is its critical pressure, Pa; ω is an acentric factor, for water vapour $\omega = 0.344$.

The saturation temperature at incoming gaseous mixture is calculated by iterating with a first approximation of compressibility factor $z_v = 1$. The partial vapour pressure P_v is determined from Eq. (7), and at this value, the saturation temperature T_s obtained from equilibrium condition of Eq. (3). At this T_s and P_v values, new iteration for compressibility factor z_v is calculated by Eq.(8) and so on until the difference of two successively calculated values of compressibility factor is not bigger than specified accuracy (0.0001) and the iterations are terminated. It requires usually not more than 3 iterating calculations.

On constructing the Hot Composite Curve, all-temperature range of exhaust gas cooling process development is divided into small equal intervals. The calculations are starting from the highest temperature by calculating the change of stream enthalpy at all successive temperature intervals. When the temperature of the gaseous mixture becomes smaller than inlet saturation temperature, the partial pressure P_{vi} in the saturated gaseous mixture at the temperature of interval end T_{mxi} is determined from the saturation condition of vapour:

$$P_{vi} = P_{sat_v}(T_{mxi}) \quad (10)$$

The mass flowrate of vapour at temperatures T_{mxi} is determined from Eq. (7) at a fixed mass flowrate of non-condensing part of gaseous mixture:

$$G_{vi} = \frac{R_g}{R_v \cdot z_{vi}} \cdot \frac{G_g \cdot P_{vi}}{(P_{mx} - P_{vi})} \quad (11)$$

On this obtained values, the enthalpy change on each temperature interval is calculated by Eq. (1), and Hot Composite Curve is finally built. After that, the set of cold streams is determined that require to be heated in that temperature range on retrofit for exhaust gas heat utilisation. The

Cold Composite Curve is constructed following the conventional PI rules (Klemeš 2013). Upon this Pinch, the position is determined, and the structure of Heat Exchanger Network (HEN) can be built following PI methods. It must also be accounting for the possibility of exhaust gas stream splitting on gas and liquid parts after some point where heat contained in a liquid phase is becoming higher than heat contained in a remaining gaseous mixture, as it is illustrated in Case Study presented in the following section.

3. Case study

The case of tobacco factory equipped with the plant for superheated steam drying of tobacco is considered. The parameters of drying plant and possible streams for heat utilisation are presented in paper by Arsenyeva et al. (2016). More detailed analysis of the heat wasted at the factory revealed that a significant amount of heat is also wasted with flue gases leaving Natural Gas (NG) fired boiler of the factory. The boiler is producing steam for drying plant but also for space heating of factory workshop, administrative and other buildings. The heat for space heating is supplied through heat exchanger by steam heating ethylene glycol solution circulating in closed circuit. The temperature of exhaust flue gases of the boiler is 180 °C. In Table 1 are presented time-averaged parameters of exhaust gases streams leaving drying plant and boiler flue gases. There are also parameters of cold streams that require heating. It is ethylene glycol of the space heating circuit and feeds water for the boiler.

The volumetric flue gas composition after NG combustion is taken as 14 % CO₂, 11.5 % H₂O, 2.8 % O₂, and 72% N₂. The exhaust gas after drying is a mixture of air and steam. The physical properties are taken from paper of Rowley et al. (2007), according to Perry's Chemical Engineers' Handbook (Perry and Green 2008). The steam saturation pressure at temperature T is determined by Eq. (12).

$$P_{vs} = \exp\left(73.649 - \frac{7258.2}{T} - 7.3037 \cdot \ln T + 4.1653 \cdot 10^{-6} \cdot T^2\right) \quad (12)$$

Fig. 1 The curves of flue gas cooling: 1 – Natural Gas; 2 – Biomass; 3 – Exhaust gas after drying plant.

Table 1 Streams data

No	Stream	Stream type	Supply temperature (t _s), °C	Target temperature (t _T), °C	Mass flowrate (G), kg/s	Heat capacity flowrate (CP), kW/°C	Heat flow (ΔH), kW
1	Flue gas after boiler	Hot	180	70	0.508	-	185.7
2	Exhaust gas after drying	Hot	140	5	0.28	-	694
	H ₂ O after drying		140	5	0.25		
3	Ethylene glycol	Cold	50	70	8.600	35	700
4	Boiler feed water	Cold	5	40	0.25	1.043	36.5

The curve of enthalpy change in the process of cooling flue gas after NG combustion is presented in Figure 1. After the NG combustion depicted by curve 1, the flue gas can release 187.7 kW of heat on cooling to 0 °C. Of this heat 73.9 kW is released as sensible heat before cooling to dew point at 49.5 °C and mostly latent heat of condensing steam is 113.8 kW or about 1.5 times higher. The calculations are made for a steam share in flue gas 11.5 % on volume. For comparison by curve 2 in Figure 1 is presented the process of cooling the same amount of flue gas with steam volumetric content 23.0 %, that can be at combustion of some coals (Dave et al. 2011) or biomass (Molcan et al. 2011). In this case, the sensible heat released before reaching dew point at 64.8 °C is 69.9 kW, but latent heat over 3.3 times higher in value of 233.8 kW and is available at higher temperatures. It shows the importance of accounting the type of fuel and steam content in flue gas when making the decision on heat utilisation.

Another source of waste heat is exhaust gas after drying process (stream 2 in Table 1) with steam content 93.6 % on volume. The enthalpy change on this gas cooling is shown by curve 3 in Figure 1. The total change of enthalpy on cooling from 140 °C to 0 °C is 695.9 kW. The sensible heat of cooling to dew point temperature 108.8 °C is 17.7 kW or only 2.5 % of the total. The main amount of heat that can be received from these exhaust gases is generated by the predominantly latent heat of vapour condensation at temperatures below the dew point. With condensation of vapour the temperature of exhaust gas is changing according to equilibrium conditions with decreasing of the driving force of heat transfer process in heat exchangers and minimal temperature approach in PI. In the case considered the total amount of heat that can be utilised from exhaust gases after drying is over 3 times bigger than heat from the flue gas after

NG combustion in the boiler. It is also having much higher levels of temperature that is making its utilisation more convenient.

To account the heat generated by both waste heat sources, the Hot Composite Curve (HCC) of the process is built, as shown in Figure 2. It is obtained by summing at the same temperatures the enthalpies in streams of flue gases of boiler fired with NG and exhaust gases from drying plant. The total amount of heat that can be obtained with the cooling of these exhaust gases to 0 °C is 884.0 kW. In Figure 2 is also presented Cold Composite Curve (CCC) calculated for cold streams 3 and 4 of Table 1. The total amount of heat required to heat all cold streams is 736.5 kW. However, some part of the heat from hot streams is available at temperatures lower than it is required by cold streams. It can be seen from Figure 2 where CCC is moved left to $\Delta T_{min} = 1$ °C The Pinch temperature for cold streams is 50 °C. The amount of heat available from hot streams being cooled to 51 °C is 710.1 kW. Cold streams require at these temperatures 700 kW. But the additional heat from utility boiler may be needed for calculations at any ΔT_{min} and for periods when drying unit is not in operation. The use of a heat pump is not considered in this paper.

Fig. 2 Composite Curves for $\Delta T_{min} = 1$ °C

According to "golden rule" for development of optimal Heat Exchanger Network (HEN) (Linnhoff et al. 1982), no heat has to be transferred across the Pinch. Below the Pinch, it is required to heat cold stream 4 (Table 1) with 36.5 kW. The available heat from hot streams is much bigger, 168 kW. It should be decided from what hot stream below the Hot streams Pinch temperature 36.5 kW of heat can be taken using a heat exchanger with minimal cost. Both hot streams have enough enthalpy to fulfil this task (see Figure 1).

With condensation of vapour, the flowrates of individual phases are considerably changing with temperature as is shown in Figure 3 for exhaust gases after the drying process with volumetric content of steam in incoming gas 93.6 %. The heat capacity flowrate of exhaust gases streams is also considerably changing with temperature, as also heat capacity flowrates of phases in two-phase condensing flow (see Figure 4).

Fig. 3 Flowrates of phases in exhaust gas stream after drying process: curve
(1) – Liquid phase; (2) - Gas-phase.

Fig. 4 Heat capacity flowrates of exhaust gas after drying process: Curve (1) – Total heat capacity flowrate; (2) - Heat capacity flowrate of gas-phase; (3) - Heat capacity flowrate of liquid

Fig. 5 Heat capacity flowrates of exhaust gas after drying process: Curve (1) – Total heat capacity flowrate; (2) - Heat capacity flowrate of gas-phase; (3) - Heat capacity flowrate of liquid

At temperature below 70 °C, more than 98 % of steam is condensed, and heat capacity flowrate of the liquid becomes predominant. The cooling of the liquid phase on 60 °C can give about 60 kW of heat that is fairly enough for heating cold streams below Pinch. The presence of gas in liquid can significantly increase the pressure drop in a heat exchanger and to avoid that the two-phase stream of exhaust gas can be split on liquid part directed to liquid-liquid heat exchanger and gas part treated separately. In the case considered it is discharged to ambient. The obtained liquid after treatment can be directed back to the boiler or used for another need of the factory that can contribute to the economy in water consumption.

The water content in flue gases of boiler is 11.5 % on volume and mass share of created condensate stream much smaller, as also its heat capacity flowrate, as shown in Figure 5. It is no reason to split this stream. In considered case there are not cold streams remains to utilise this heat below Pinch and it should be discharged to the atmosphere.

With the use of Composite Curves in Figure 2 and accounting for the discussed features of exhaust gases cooling process the structure of HEN Grid Diagram shown in Figure 6 is obtained following the version presented by Smith (2005). However, there has also been some more recent development of an Extended Grid Diagram (Yong et al. 2015) and Shifted Retrofit Thermodynamic grid Diagram (Wang et al. 2020a), which offer several even more enhanced features.

This structure is not changing with the variation of ΔT_{min} that can be optimised with an economic objective function. The optimal ΔT_{min} is also determining the Hot Pinch temperature. At this temperature, the stream 2 is split on gas and condensed liquid flows.

Fig. 6 The Grid Diagram of HEN for heat utilisation at $\Delta T_{min} = 23$ °C

Accounting for the nature of heat exchanging streams and working temperatures for heat recuperation at different HEN matches the specific types of PHEs are selected. For cooling of exhaust air from drying plant, the plate-and-frame PHE TS6M produced by AlfaLaval (Alfa Laval 2020) is used. The plates of this PHE have corrugations with rather big height about 4 mm

compare others PHE plates with height 2-3 mm. It allows having a bigger cross-section area of channels to decrease pressure drop of heat exchanging stream, that is important for gaseous streams and condensing duty applications. The calculations of steam-air condensation in this PHE were made using the mathematical model presented in paper by Kapustenko et al. (2020) with heat and mass transfer coefficients determined according to Arsenyeva et al. (2014). The results are presented in Table 2 with the purchasing price in € estimated by Equation (Arsenyeva et al. 2016):

$$Pc_{HE1}=978 \cdot Fa_{HE1} + 3,100 \quad (13)$$

For liquid-liquid duty on a position HE2 AlfaLaval plate-and-frame PHE of type M3 is selected with plates having corrugation height 2mm, that is efficient for liquid heat transfer at small channel length. The calculations are made according to the method described in the paper by Arsenyeva et al. (2009). The results of heat transfer area calculations are also presented in Table 2 with the price estimated according to Equation from paper (Arsenyeva et al. 2016).

$$Pc_{HE2}=644 \cdot Fa_{HE2} + 1,216 \quad (14)$$

Table 2 The results of PHEs designs at varied ΔT_{min}

N	Parameter, units	Values								
1	Temperature of gas after HE1, °C	55	58	63	68	73	78	83	88	93
2	HE1 area, m ²	61.2	42.7	19.3	9.1	6.4	4.6	3.4	2.7	2.2
3	HE1 temperature of liquid, °C	53.1	55.3	58.6	61.8	65.2	68.8	72.7	77.1	81.8
4	HE1 liquid flow rate, on exit, kg/s	0.249	0.248	0.247	0.246	0.245	0.244	0.241	0.237	0.231
5	HE1 heat load, kW	625.3	623.4	618.1	612.4	605.8	597.8	587.9	574.9	555.8
6	HE2 area, m ²	1.61	1.18	0.86	0.74	0.58	0.48	0.45	0.42	0.38
7	HE2 heat load, kW	36.5	36.5	36.5	36.5	36.5	36.5	36.5	36.5	36.5
8	HE3 area, m ²	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4
9	HE3 heat load, kW	50.1	50.1	50.2	50.3	50.4	50.5	50.6	50.7	50.8
10	Total heat load, kW	711.9	710.0	704.8	699.2	692.7	684.8	675.0	662.1	643.1
11	Heat leaving with gas phase, kW	32.4	34.3	39.5	45.2	51.9	59.8	69.7	82.9	101.8
12	ΔT_{min} , K	5.0	8.0	13.0	18.0	23.0	28.0	33.0	38.0	43.0

Stream 1 of flue gases has an initial temperature of 180 °C. It is out of working range for rubber gaskets of plate-and-frame PHE. Here the heat exchange between gaseous and liquid streams must be performed. For cooling of flue gases at position HE3 the welded pillow-plate heat exchanger (PPHE) is chosen to count for high temperature and demand to accommodate big gas flowrates that are directed in a channel between pillow panels (Arsenyeva et al. 2019). In Table 2 are presented the results of calculations of the PPHE heat transfer area obtained according to the method described in the paper of Arsenyeva et al. (2019). The price is estimated according to Equation for welded PHEs from the book by Klemeš et al. (2015):

$$Pc_{HE3} = 4,690 \cdot Fa_{HE3}^{0.7} + 4,280 \quad (15)$$

The cost of equipment installation IC= 41,170 € and cost of project development PC=6,060 € are taken based on data for real expenses on implementation at the factory the pilot substation for heat utilisation from exhaust gases of drying plant reported in the paper of Gariev et al. (2014). The purchasing cost of the equipment:

$$PEC = Pc_{HE1} + Pc_{HE2} + Pc_{HE3} \quad (16)$$

The total capital investment cost:

$$TIC = IC + PEC + PC \quad (17)$$

The annualised capital investment cost is determined according to Eq. (18), proposed in the book by Smith (2005):

$$AIC = TIC \cdot \frac{i \cdot (1+i)^n}{(1+i)^n - 1} \quad (18)$$

where i is the yearly interest rate for capital borrowing and n is time in years for the borrowing of the capital (it is assumed $i=0.10$ and $n=3$).

The price of steam is taken as 380 € · kW⁻¹ · y⁻¹ and on its operational cost OC is determined assuming that the installed equipment is operating only during the heating season or half a year.

The total annual cost:

$$TAC = AIC + OC \quad (19)$$

The influence of ΔT_{min} on total annual cost, annualised capital investment and operational cost are shown in Figure 7. The minimum of TAC is observed at ΔT_{min} equal to 23 °C. In this case, the amount of 692.7 kW of heat is saved and would not need to be produced by the boiler for

space heating. The cost of this saved energy in the form of steam in the half-year period of the heating season is 132,000 €. The capital investment for this $\Delta T_{\min}$ option is 71,070 € or about 1.85 times lower and payback period for the proposed waste heat utilisation system is about 3.2 months after the installation was commissioned.

Fig. 7 The change of economic characteristics with $\Delta T_{\min}$:

The proposed system for heat utilisation from exhaust gases in 182 days of operation during heating season is saving about 10.9 TJ/y of heat produced by combustion of natural gas. According to an analysis presented by Juhrich (2016), the carbon dioxide emission factor for such NG is about 55.16 t/TJ and the implementation of the proposed system is leading to reduction of about 600 t/y of CO₂ emissions, as also accompanying SO_x and NO_x emissions produced with NG burning. The flowrate of condensed water from exhaust gases after drying plant at $\Delta T_{\min}$ equal to 23 °C is 0.245 kg/s. It means that 3,850 t/y of water is not emitted into the environment and after some treatment can be returned to a steam-generating boiler that is leading for saving of that amount of freshwater.

Conclusions

The exhaust gases discharged into the environment after a number of industrial processes contain a considerable amount of energy in the form of sensible and latent heat that is wasted if not properly utilised. The efficient utilisation of this heat and its integration into the existing energy system of the enterprise is possible with the application of Process Integration methodology and the use of compact plate heat exchangers with enhanced heat transfer. The accurate estimation of heat available for utilisation from exhaust gases at different temperature levels requires the construction of Hot Composite Curve accounting the sensible and latent heat that can be extracted in the process of exhaust gases streams cooling. The method of Composite Curve construction for exhaust gases with different content of the condensable vapour part is proposed. It is accounting for the nonlinear character of stream enthalpy relation to temperature in cooling processes involving phase change of condensable component. The use of Process Integration allows identifying the structure of HEN accounting for streams requiring heating in the energy system of the enterprise. The efficient realisation of the determined HEN requires to consider the correct choice of heat exchanger type for each position depending on the streams nature and process parameters. It is preferably compact plate heat exchangers with accounting for the possibility of condensing streams splitting at some temperature on gas and liquid phases for more efficient heat exchange in individual heat exchangers. The final selection of PHEs on

different positions in HEN is made with economy optimisation on Total Annual Cost as optimising criterion. The software on PC for method realisation is developed.

The method is illustrated with presented Case Study considering the utilisation of heat from two streams of exhaust gases existing at one enterprise. There is the stream of exhaust gas after the process of superheated steam tobacco drying with steam content 93.6 % on volume and stream of flue gas after natural gas-fired boiler with volumetric steam content 11.5 %. It shows the capabilities of PI application for utilisation of waste heat from exhaust gases with the use of efficient PHEs selected according to their position in heat utilisation HEN. The implementation of developed HEN allows saving about 10.9 TJ/y of heat energy. It means the reduction of NG consumption and corresponding CO₂ emissions on 600 t/y. About 3,830 t of water steam is not discharged to the atmosphere and as water can be returned to the production process.

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Figures

Figure 1

The curves of flue gas cooling: 1 – Natural Gas; 2 – Biomass; 3 – Exhaust gas after drying plant.

Figure 2

Composite Curves for $\Delta T_{min} = 1\text{ }^{\circ}\text{C}$

Figure 3

Flowrates of phases in exhaust gas stream after drying process: curve (1) – Liquid phase; (2) - Gas-phase.

Figure 4

Heat capacity flowrates of exhaust gas after drying process: Curve (1) – Total heat capacity flowrate; (2) - Heat capacity flowrate of gas-phase; (3) - Heat capacity flowrate of liquid

Figure 5

Heat capacity flowrates of exhaust gas after drying process: Curve (1) – Total heat capacity flowrate; (2) - Heat capacity flowrate of gas-phase; (3) - Heat capacity flowrate of liquid

Figure 6

The Grid Diagram of HEN for heat utilisation at $\Delta T_{min} = 23 \text{ }^{\circ}\text{C}$

Figure 7

The change of economic characteristics with ΔT_{min}

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