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EFFICIENT HEAT RECOVERY FROM HYDROGEN AND NATURAL GAS BLEND COMBUSTION PRODUCTS

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The introduction of hydrogen and natural gas blends in existing gas transportation and distribution networks would ensure faster and more efficient decarbonization of energy sector, but, at the same time, this process would request solution of many practical and technical problems. This paper explores thermodynamics of hydrogen and natural gas blend combustion products and estimates the amount of condensate and latent energy recovery from flue gas as a function of condensing temperature. The efficient energy recovery depends on network return temperature, and it is possible to overcome this limitation by implementation of heat pump for extraction of low temperature heat from flue gases. The case study considers operation of heat only boiler and flue gas condenser with integrated cascade of heat pumps, which consist of absorption lithium bromide-water chiller (in heat pump mode) and vapour compression unit. Presented results of energy recovery hence are limited by data collected from the natural gas combustion for district heating network energy supply. However, previous thermodynamic consideration allows extending the obtained results for case of hydrogen and natural gas blend combustion. A proof of concept of heat recovery by combination of flue gas condenser supported by a cascade of heat pumps demonstrates the efficiency in case of hydrogen and natural gas blend combustion.

Keywords: *Cascade utilization, condensation recovery, heat pumps, hydrogen, natural gas.*

1. INTRODUCTION

Green hydrogen presents a viable alternative to fossil fuel and ensures sustainable energy transition and decarbonization of many sectors of national economies. This can be achieved in line with improvement of air quality and expansion of energy storage options. Use of hydrogen in the energy sector has a long history, which begun with the first internal combustion engines over two centuries ago. Currently, it forms an integral part of the modern chemical and refining industry [1].

Hydrogen offers many very valuable features as a fuel; it is energy-dense when in its liquefied form, has a wide flammability range, and can be manufactured via a variety of ways. In some applications, it can be substituted in a relatively straightforward way as an alternative fuel, displacing natural gas, gasoline, kerosene and diesel in combustion applications [2]. Hydrogen has potential to fill certain energy supply requirements that may not be easily met using battery electric storage, for example, for gas boilers in the home, as a fuel for heavy-duty vehicles, off-road machinery, trains, back-up capacity for grid electrical supply and as a fuel for global shipping and aviation [3]. The preeminent argument in favour of the use of hydrogen as a fuel is that the major waste product formed from its use is water vapour.

In order to support further spread of hydrogen in different industries, more than thirty countries worldwide already have developed the national framework documents – strategies and roadmaps, for promotion of hydrogen as fuel and energy carrier. As for Europe, the European Hydrogen Backbone project initiates the creation of continent-wide hydrogen transportation and storage infrastructure [4].

The power systems of the Baltic States carry significant expansion potential in renewable energy sources (hereinafter – RES) based generation, and by 2050 they could generate a total of 21.34 tera-watt hour (hereinafter – TWh) of electricity from RES. This corresponds to 72 % of the total regional electricity consumption [5]. The mature technology of water electrolysis would allow for smooth integration of RES into green hydrogen production with a purpose to balance energy generation and consumption.

To replace hydrocarbons in electricity generation, the green hydrogen can be used instead, especially in power plants, which can operate in a cogeneration mode and provide both electricity and heating energy for district heating.

The fuel cell needs only pure hydrogen of high-grade quality for operation, what can be a certain limitation in current circumstances of limited green hydrogen availability. Combustion of hydrogen is less efficient than fuel cell conversion to electricity, but it significantly reduces requirements for quality of fuel and opens a way to extensive use of different gaseous fuel blends like one of the natural gas and hydrogen, biomethane and hydrogen, propane and hydrogen, etc.

The transition to blend of hydrogen and natural gas and/or biomethane would allow continuing exploitation of existing natural gas boilers and burners, not replacing them immediately with new ones. In addition, combustion of such blend would contribute to greenhouse gas (hereinafter – GHG) emission reduction targets by gradual decarbonization of the heat energy sector, which would be achieved just by an increase of hydrogen share in hydrogen and natural gas blend alone. With moder-

ate content of hydrogen (if less than 20 %), it allows for further use of existing natural gas storages, pipelines and other distribution system elements without fundamental investments and technical changes [6]–[8].

However, there are several concerns regarding wider use of hydrogen in low-carbon heating. First, hydrogen has low ignition energy and low density; therefore, it should be stored under high pressure. This can create explosion risks, plus small size of hydrogen molecules can ensure high capability to escape from storage tanks, pipes through all kind of joints. They can also penetrate through many hydrogen-non-resistant materials, causing embrittlement and creating the so-called hydrogen assisted cracking in various metals and their alloys [9].

Second, concerns would derive from potential health risks presented by increased nitrogen oxide (hereinafter – NO_x) emissions during combustion of hydrogen and natural gas blend in comparison with combustion of natural gas or biomethane. Experimental study of hydrogen and natural gas blend combustion demonstrates a minor increase of NO_x (only natural gas f=0) from 45 to 52 ppm for f=30 % [8]. Another research presents the following results: when hydrogen is added to natural gas by 25 %, 50 % and 75 %, the NO_x emission level increases by 92.81 %, 219.72 % and 360 % (485 ppm), respectively [10].

However, both factors of concern can be avoided by means of engineering solutions: correct materials selection and use of low NO_x burner.

Hydrogen combustion does not emit GHG, but origin of hydrogen or way it is produced can be a challenge for low-carbon heating. Steam Methane Reformation (hereinafter – SMR) currently provides the lowest cost of produced hydrogen, but this process releases significant amount of carbon dioxide (hereinafter – CO_2) into atmosphere. Also, due to limited efficiency of SMR, which is about 70 %, it is responsible for bigger CO_2 emission per unit of energy produced than natural gas combustion process. An alternative way to produce hydrogen – by means of water electrolysis and RES, still is more costly. Taking in account a need for RES energy storage and balancing, it might support introduction and wider use of hydrogen and natural gas blend.

Combustion product of hydrogen and natural gas blend will contain more water steam, when hydrogen share in fuel will grow. Efficient heat recovery from flue gases would be possible, if combustion products would be cooled down lower than dew point and water steam would condensate and release latent heat. If we condensate all water steam from flue gases, it would be possible to shift from a low heating value (hereinafter – LHV) to a high heating value (hereinafter – HHV) and save a significant amount of fuel consumed: from 10 % in case of natural gas, and up to 18 % in case of hydrogen (f=1). Return flow temperature in the heating system would be a limitation for efficient moisture condensation from flue gas. Application of heat pump for heat recovery from flue gas would open a way to overcome limitation of this kind. Operation of HOB with FGC and AHP (water steam fired chiller-heater) demonstrates an increase in natural gas combustion output from 70 by 6 megawatt (hereinafter – MW) (sensitive heat of flue gas recovery in FGC) plus 2 MW extracted by AHP (latent heat of moisture from fly gas) [11]. The share of AHP in heat recovery limited by 2/76 = 2.6% of total HOB power in the considered case study.

Nevertheless, there are many theoretical and practical studies on application of heat pumps and heat pump cascades with the purpose to extract more energy from

flue gas [12]–[14]. Issue of temperature lift is very significant for energy recovery potential increase, and the combination of absorption and compression heat pumps becomes relevant for hydrogen and natural gas blend combustion, since water steam condensing will get big share in energy balance. This paper presents a case study of the natural gas boiler, where already installed AHP with FGC with addition of vapour comparison pump is targeted to extract more sensitive heat from condensate of FGC. Available operational data allow performing estimation on heat recovery gain in case of shift from natural gas to hydrogen and natural gas blend.

2. METHODOLOGY

Chemical composition of hydrogen and natural gas blend and design of burner affect combustion product properties. The main parameters for combustion are hydrogen fraction f and amount of excess air λ. Suggested approach will follow [7] and reduce hydrogen and natural gas blend composition to methane-hydrogen blend, with a purpose to reveal how dew point temperature in combustion product depends on input parameters. The aim of this consideration is to establish relation between actual cooling (condensing) temperature and blend LHV.

2.1. Thermodynamic Preliminaries

The methane-hydrogen blend combustion can be represented by following chemical equation with an aim to establish mass and heat balance for reaction on base of kilomole

$$fH_2 + (1\text{-}f)CH_4 + \lambda(2-1.5f)(0_2 + 3.76N_2) \rightarrow \\ \rightarrow (1-f)CO_2 + (2-f)H_2O + 3{,}76\lambda(2-1.5f)N_2 + (\lambda-1)(2-1.5f)O_2. \quad (1)$$

Partial water pressure and respectively dew point temperature can be evaluated as

$$\frac{P_{H20}}{P} = \frac{2-f}{1-0.5f+9.52\lambda-7.14f\lambda}. \quad (2)$$

For a hydrogen and natural gas blend combustion product (dry) kmol number,

$$\mathrm{N_{mix}} = 0.5\mathrm{f} + 9.53\lambda - 7.14\mathrm{f}\lambda - 1. \quad (3)$$

Mass of dry combustion product in kg for one kmol of hydrogen and natural gas blend

$$m_{dc} = 4f + 274\lambda - 206\,\lambda f - 20. \quad (4)$$

Molecular mass of dry combustion product

$$M_{cp} = \frac{4f + 274\lambda - 206\,\lambda f - 20}{0.5f + 9.53\lambda - 7.14f\lambda - 1}. \quad (5)$$

For a humidity ratio of combustion product (kg of water steam per kg of dry combustion product), one can show that

$$d = \frac{P_{H20}}{(P-P_{H20})}\frac{M_{H20}}{M_{CP}} = \frac{1}{(\frac{P}{P_{H20}} - 1)}\frac{18}{\left(\frac{4f + 274\lambda - 206\,\lambda f - 20}{0.5f + 9.53\lambda - 7.14f\lambda - 1}\right)}. \quad (6)$$

Water steam amount in flue gas after combustion 1 kmol of hydrogen and natural gas blend

$$D = d \cdot m_{dc} = \frac{18(0.5f + 9.53\lambda - 7.14f\lambda - 1)}{\frac{P}{P_{H20}} - 1}. \quad (7)$$

However, amount of water steam in kg already known from chemical reaction

$$D = 18 \cdot (2 - f). \quad (8)$$

If flue gas is treated in condenser (cool down to temperature t, which is lower than dew point) and it becomes possible to calculate amount of condensate per kmol of hydrogen and natural gas blend

$$D_c = 18 \cdot (2 - f) - d \cdot m_{dc} = 18\left(2 - f - \frac{(0.5f + 9.53\lambda - 7.14f\lambda - 1)}{\frac{P}{P_{H20}} - 1}\right). \quad (9)$$

Water steam saturated pressure can be described by number of empirical equations: Antoine, Tetens, Arden Buck, Goff-Gratch equations etc. In the present paper we use Tetens formula as it is simple and accurate (in range of 10 C to 100 C deviation from table value is within 0.5 %):

$$P_{H20} = 0{,}0061078\exp\left(\frac{17.27t}{t + 237.3}\right), bar \quad (10)$$

whereupon Eq. (10) may be used in a reversed way

$$\frac{1}{\frac{P}{P_{H20}} - 1} = \frac{1}{P \cdot 163.72\, exp\left(-\frac{17.27t}{t + 237.3}\right) - 1}. \quad (11)$$

Hydrogen and natural gas blend LHV in kWh/ kmol of fuel (hydrogen LHV = 67.2 kWh/kmol):

$$Q_{HENG} = LHV_{NG}(1 - f) + 67.2f. \quad (12)$$

To express the ratio of released heat Q (combustion + water steam condensation) to LHW of hydrogen and natural gas blend as a function of flue gas cooling temperature t

$$\frac{Q}{Q_{HENG}} = 1 + \frac{18 \cdot L}{LHV_{NG}(1 - f) + 67.2f}\left(2 - f - \frac{0.5f + 9.53\lambda - 7.14f\lambda - 1}{P \cdot 163.72\, exp\left(-\frac{17.27t}{t + 237.3}\right) - 1}\right). \quad (13)$$

Taking into account that natural gas had no permanent composition, the present paper will consider hydrogen and natural gas blend as hydrogen / methane blends. The value of LHV for methane will be 222.8 kWh/kmol and value 0.69 kWh/kg will be used for latent heat L. This simplification will not reduce the value of obtained equation (12), (13), but allow yielding an alternative fitting function:

$$\frac{Q}{Q_{HENG}} = 1 + \frac{12.5}{222.8 - 155.6f}\left(2 - f - \frac{0.5f + 9.53\lambda\ - 7.14f\lambda - 1}{P \cdot 163.72\, exp\left(-\frac{17.27t}{t + 237.3}\right) - 1}\right), \qquad (14)$$

where P is total pressure of gas blend in bar.

2.2. Performance Comparison

The presented thermodynamic consideration is based on gas blend description and involves empirical equation for saturation temperature definition. Obtained equations allow finding an answer how f blend composition and λ amount of excess air influence dew point temperature. Numerical consideration presented in Fig. 1 and Fig. 2 is performed for hydrogen-methane blend. However, derived equation can be used to calculate properties for hydrogen and natural gas blend by implementation of the natural gas composition.

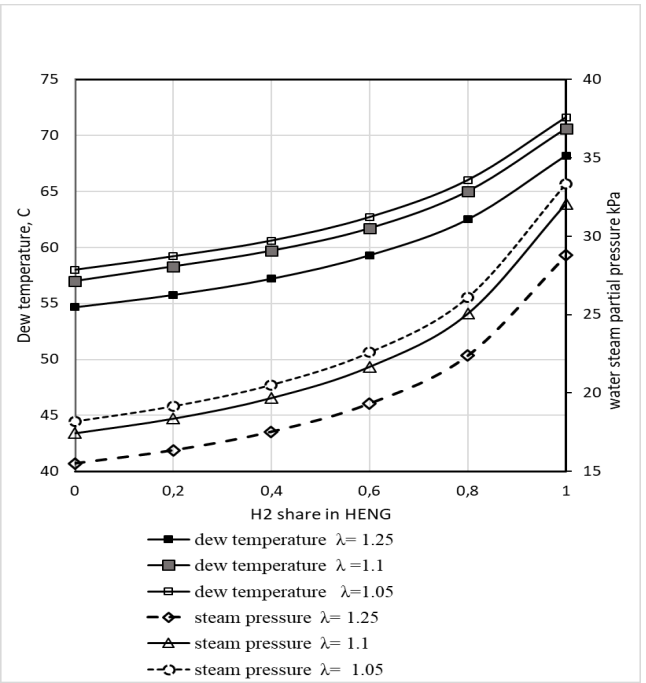


Fig. 1. Dew temperature and water steam saturated pressure vs methane-hydrogen blend composition.

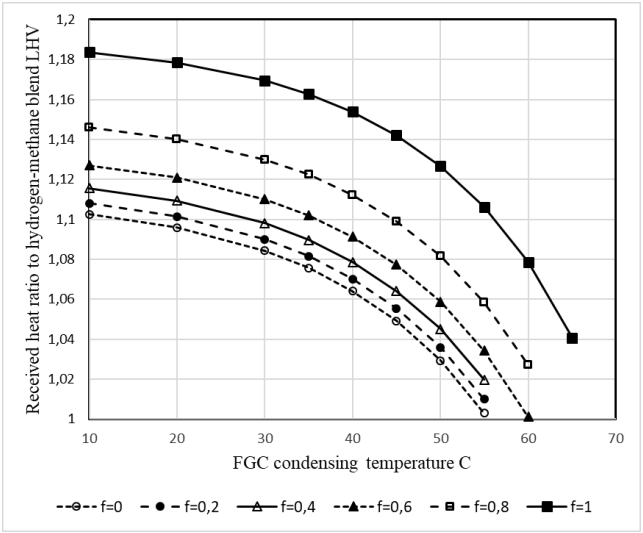


Fig. 2. Received heat ratio to methane-hydrogen blend LHW for different composition f and various condensing temperature t (when λ=1.2).

Figure 1 presents saturation pressure of water steam for various f and λ in range of 1.05 to 1.25 according to Eq. (2), then corresponding value of saturation temperature is obtained from steam-table and is designated as dew point temperature. The dew point temperature increases with f growing, so, when hydrogen and natural gas blend has larger share of hydrogen, dew point can be up to 65 to 72 ^{0}C depending on λ. When λ increases, the value of saturation pressure and dew point decrease. This value can be important for FGC design in case of hydrogen and natural gas blend with a different hydrogen share.

Moisture condensation from flue gas can provide heat recovery and increase energy output, but still HOB efficiency is a function of many design parameters like insulation, air temperature etc. Consideration presented in Eq. (14) was limited by t – cooling temperature, f – composition of blend and λ. Results of calculation presented in Fig. 2 and obtained for λ=1.2 demonstrate that cooling temperature t=55 ^{0}C for pure methane (f=0) will produce no recovery, because the temperature is around dew point and condensation is not possible, but for f=1 (pure hydrogen) energy recovery will be around 10 %.

This can serve as evidence that flue gas condensation from combustion product of hydrogen and natural gas blend with high hydrogen amount will be less sensitive to t2 network return temperature. However, Fig. 2 reveals that, if cooling temperature in FGC is as low as 20 ^{0}C (much less than T2) heat recovery will increase from 9 % (f=0) to 18 % (f=1).

Additionally, it is possible to investigate influence of λ on heat recovery output. The analysis represented in Fig. 2 indicates that flue gas condensation at lower temperature will significantly increase amount of heat recovery. Developed equations (1)–(14) suggested for purpose to estimate the desired level of t temperature of condensation since it would be lower than T2. Decrease of t would require more energy input in the already discussed heat pump with different driving energy (thermal for absorption and electrical for compression type of units).

Application of low temperature lift heat

pumps would enhance heat recovery and definitely decrease fuel consumption, but it is important to look at total energy balance. Therefore, all considerations presented before should be reinforced by real life operation results, viewed from total energy efficiency standpoint. At the moment, hydrogen and natural gas blend is not yet a common fuel, and it is good chance to foresee performance of FGC supported by various combinations of heat pumps.

3. CASE STUDY

The performance of natural gas fired HOB boiler (rated power 100 MW) equipped with flue gas condenser and additional cascade of heat pumps is considered with purpose to estimate moisture condensation from flue gas in real operational condition. The performance data are available only for natural gas combustion, but since previous consideration it will be possible to collect data of heat recovery estimation when it will be hydrogen and natural gas blend combustion.

3.1. Experimental Setup

The considered HOB supply water flowed in heating network at temperature T1, the return flow from network had temperature T2. The value of T1 followed ambient temperature for outside weather (in range of 10 to -15 ^{0}C) direct temperature was around 100 ^{0}C (outside -15 ^{0}C), when outside temperature became around zero, direct temperature decreased toT1= 70 ^{0}C. The network return temperature T2 had corresponding range from 50 ^{0}C to 40 ^{0}C.

Water flow from return line had temperature T2 and this was sufficient to cool down flue gas from boiler TF1 =140 ^{0}C.

However, in fact it is difficult to ensure efficient moisture condensation from flue gas without AHP. Temperature difference between flue gas flow and water in FGC can be at least 10–15 ^{0}C and more; therefore, flue gas is left with temperature above dew point (Fig. 2). The retrofit target of available FGC was to decrease temperature in flue gas: by circulation loop 6 was connected to AHP evaporator E. This ensured deeper heat recovery from flue gas: temperature after E was around 20 ^{0}C and it grew in FGC up to 35 ^{0}C (left FGC and enter to E), thus decreasing flue gas temperature after FGC to 40–45 ^{0}C, which was not possible without application of AHP. The experiment was performed for HOB power output range from 30 to 70 MW (depending from network energy demand). Amount of condensate from FGC was limited to 3–4 m^3/h and did not grow proportionally of HOB power output. It seems that during FGC retrofit additional loop 6 faced limited contact surface with flue gas and this was a shortcoming for efficient moisture condensation from combustion product. AHP (water-fired chiller-heater direct line from HOB at 130 ^{0}C) generator G took about 2.5 MW to lift return temperature of network from T2 =40 ^{0}C up to 54–55 ^{0}C accounted as 4 MW (obtained in absorber A and condenser C).

As a result, the operation of HOB at 40 MW accounted the following: the consumption of G about 2.5 MW (hot water from HOB) was complemented by 2 MW from E by latent heat of fly gas moisture condensation (liquid about 3 m^3/h) obtained in cooling loop 6 (additional condensing part in FGC). Furthermore, the AHP performance reached COP = (2+2.5)/2.5 = 1.8 that was according

to data sheet of AHP. Since the condensate after FGC had temperature TC1 = 45 ^{0}C and was equal to flue gas TF2, it was not possible to recover this energy by warming of return flow from network. Therefore, implementation of vapour compression heat pump allowed performing condensate sensitive heat recovery.

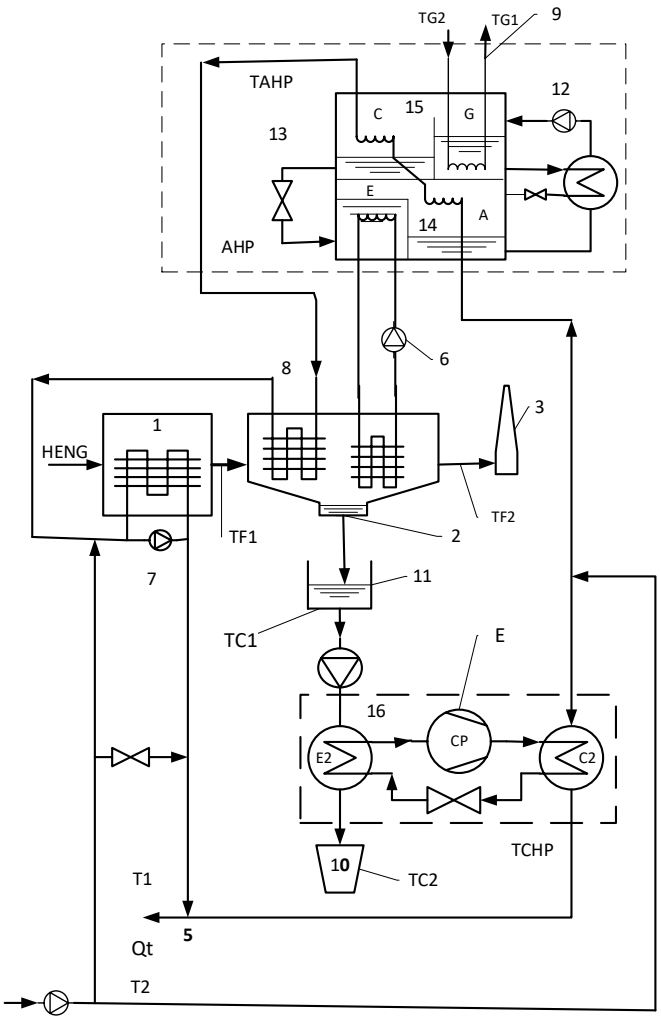


Fig. 3. Absorption and vapour compression heat pumps with flue gas condenser operation for energy recovery: 1 – Heat only boiler (HOB), 2 – flue gas condenser (FGC), 3 – chimney, 4 – network circulation pump, return water from district heating network (DHN), 5 – direct water to DHN, 6 – flue gas energy condensing loop for AHP evaporator (E), 7 – HOB recirculation pump, 8 – flue gas energy loop, 9 – hot water flow for generator of AHP, 10 – drain of flue gas condensate, 11 – condensate tank after FGC, 12 – weak solution circulation pump, 13 – absorption heat pump (AHP) , 14 – evaporator (E) and absorber (A), 15 – condenser (C) and generator (G), 16 – vapour compression heat pump, E2 evaporator (low pressure), CP refrigerant compressor VFD and C2 high pressure condenser, T1 – direct and T2 – return DHN temperature, TG1 and TG2 – generator direct and return, TC1 and TC2 condensate temperature after first and second peat pumps step. TF1 and Tf2 – fly gas temperature after boiler and fly gas condenser. TCHP – temperature after compression heat pump. EL – electrical energy consumption from compression heat pump drive.

The HOB fly gas condensate in amount of 3.5 m^3/h cooling from given TC1 to TC2=15 ^0C (flow through evaporator of CHP) produces recovery of 116 kW. The COP= 3.9 requests El=40 kW electrical power consumption by CHP compressor. In this case, return water obtained about 156 kW (40+116) and left condenser of CHP at 55 ^0C.

Based on these results, it has been shown that the fly gas heat recovery (moisture condensation) can provide about 5 % surplus energy by AHP application and CHP by condensate cooling can return additional 0.28 % to HOB power.

3.2. Experimental Findings and Discussion

This paper deals with investigation of practical way how to get more from fuel LHV by moving toward to HHV, which becomes possible only in case of full moisture condensation. Implementation of condensing gas boiler for domestic heat supply demonstrates high performance due to reason that return heating water temperature in a well-insulated house can be 30 ^0C and even lower. However, there is still in operation a large centralized heat network and issue of its decarbonization becomes acute. The return temperature in such a network is hardly possible to drop to the desired level. The motion toward fuel decarbonization includes hydrogen and natural gas blend introduction and this will increase potential for energy recovery by moisture latent heat return in the network.

The experimental set operation demonstrates that HOB follows network demand and flue gas flow rate will increase when outside temperature drops, but return temperature from network increases. This is counterforce to complete recovery of moisture latent heat. The application of heat pump provides an option to break direct influence of return water temperature on heat recovery.

The result of HOB recovery reveals that only part of all moisture can condensate in FGC. It is important to understand full recovery potential by accounting all water steam in flue gas in kg per kWh of combusted hydrogen and natural gas blend for single kmol of fuel:

$$N_{H20} = \frac{18(2-f)}{222.8-155.6f}, \frac{kg}{kWh}/kmol. \quad (15)$$

The equation gives water condensate 6.46 t/h for combustion of 40 MW (by LHV) natural gas like it was in the experimental test. If we make shift to hydrogen and natural gas blend with f = 0.5 for this power output of moisture becomes 7.5 t/h, but for pure hydrogen f=1 moisture will account to 10.8 t/h.

The whole available moisture in flue gas is possible to use for energy recovery, if temperature gradient between cooling loop and flue gas is abated. Therefore, sizing of cooling loop for FGC design had to be considered for efficient condensation with sufficient output in the whole range of HOB operation.

Finding of brake even point for energy injection in heat pumps to increase recovery extraction is beyond the specified scope of the paper, which is aimed exclusively at hydrogen and natural gas blend properties and FGC parameters in theoretical and practical applications.

4. CONCLUSIONS

Thermodynamic consideration described in the paper leads to two important findings, namely, how:

- hydrogen and natural gas blend composition would influence dew point, and
- to estimate condensate and heat recovery for various hydrogen and natural gas blends by temperature of condensation.

Along with hydrogen share in hydrogen and natural gas blend, air excess rate was involved in theoretical consideration. Therefore, the obtained relationship can be adapted for real combustion/condensation systems description taking into account real fuel composition and available range of air mixing rate.

The experimental setup data prove the concept of recovery enlargement by implementation of temperature lift in heat pump connected to a condensing device. Numerical comparison shows good compatibility of derived equation and actual condensate flow for the obtained heat recovery.

The practical value of the presented study consists in HOB and FGC operation inside outlook that reveals a weak point in the design of FGC. Limited surface of fly gas condenser halved available recovery for HOB operation on 40 % from rated output. Further increase of HOB power was not supported by recovery power grow. Thus, capacity of AHP should be balanced by ability of FGC to ensure condensation at accepted temperature gradient.

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