Mathematical model of the industrial kitchen steam condenser

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Abstract

The numerical simulation of a top hood steam condenser (*THSC*) is reported in the present thesis. A *THSC* supports the daily work of a large-scale mass cooking oven, and its main goal is to prevent the accumulation of water vapour in the air released to the kitchen. A secondary goal of the *THSC* is to reduce the unpleasant smells and to avoid increased humidity that may lead to mist appearance. The operation of a *THSC* depends on its geometry and working conditions: temperature and humidity of the air in the kitchen, flow rate of steam flowing from the oven to the *THSC*. The main goal of the present study is to develop a mathematical model to simulate the behaviour of the steam condenser implemented in Visual Basic for Application. It is based on the combination of a single pipe CFD model and on global mass and energy balances for the *THSC*. The predictions are validated against available experimental data. The developed *THSC* model requires much less computing time and human effort to produce satisfactory solutions in comparison with a fully developed CFD model and allows the user to investigate the behaviour of the *THSC* under various operating conditions, and to perform an analysis of the effect of possible configuration changes. Among the studied modifications, the reduction of the number of pipes, which has no impact on the condensation efficiency, is recommended. This improvement minimizes the cost of the *THSC* and can be carried out with presently used ovens.

Keywords: top hood steam condenser, Fluent, VBA, CFD simulation, working conditions

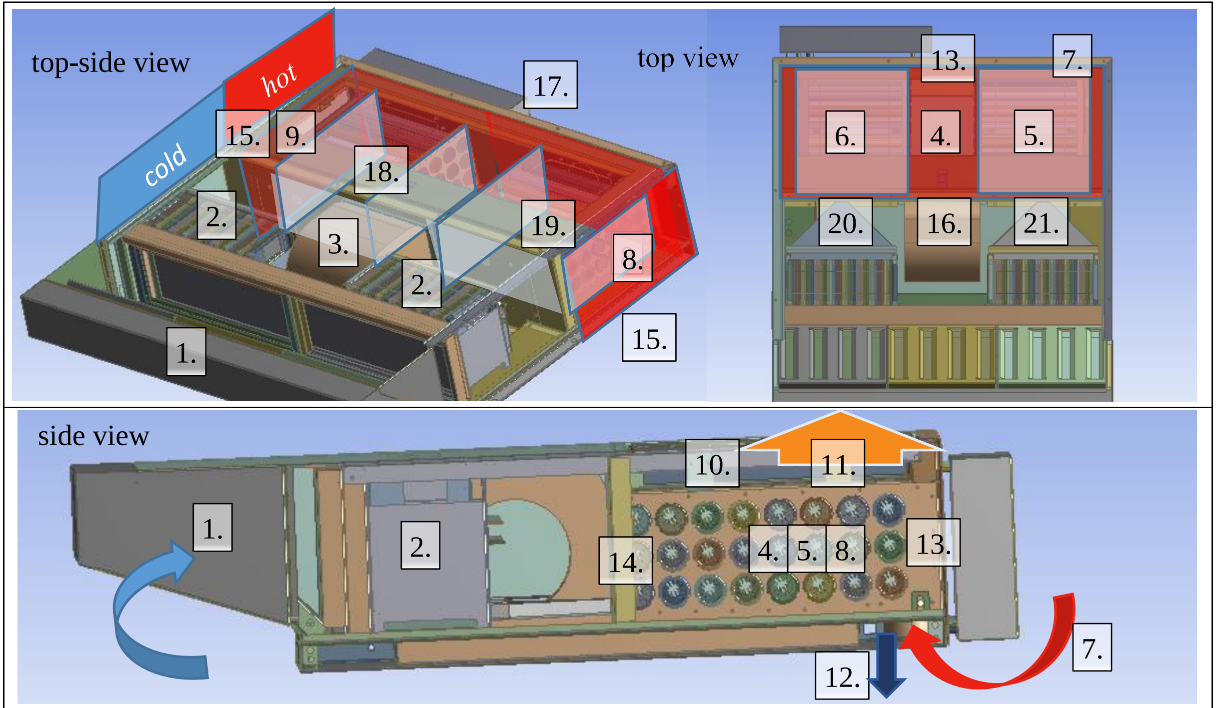
1. Introduction

A condenser is a device designed to change the phase of a working fluid from vapour to liquid, during the condensation process [1]. It is rather used among other subparts of the technological cycle than separately. Condensers are developed and exist in a great amount of non-power producing appliances. One of them, named *top hood steam condenser (THSC),* is the crucial element of the ovens, both in a kitchen and at industrial level, where various types of food are prepared [4]. Thanks to it the air outside the oven is free of unhealthy, too high humidity and fog, which may appear outside in the result of food preparation process. Since the operating conditions of the *THSC* depend on varying conditions inside the oven as well as in the kitchen, the essential step is to analyze its work. The analyzed condenser is a unique construction - the air flows inside the internally finned tubes which are surrounded by the steam. Neither theoretical references nor modern experimental study [9] considers situation in which air-cooled condenser consists of internally finned tubes. The lack of corresponding examples in literature results in two main conclusions. Unfortunately, results and conclusions from this thesis cannot be easily compared to work of other authors which makes it more complicated. On the other hand, such a situation makes this paper a real research work which is more challenging task. Afterthoughts are unique at the moment of document creation which makes them even more attractive. Moreover, the pipe of such a construction is rarely met on the market which makes the company strongly dependent on one of the subcontractors. Having in mind the condenser is already on sale and the pipe is an underbelly of a whole device, Retech company is highly interested in improvement of *THSC* production process. According to relatively big number of pipes mounted in *THSC* the suspicion appears that it is over-scaled and that the amount of pipes may be reduced. If so then minimizing the number of pipes in each condenser would lead to e.g. decreased pipe's stock and improved accounting liquidity of the company. For existing *THSC*, creation of the model constitutes the grounds for improvements and cost decrease. The Retech company declares an interest in model which can present the performance of *THSC* under various working conditions with respect to actual geometry of their device. In addition, the model is aimed to foresee trends of *THSC* behavior for structural changes. It is pointed out that crucial is the calculation time and availability of the software for the company employees. Therefore, to address all requirements a mathematical model implemented in Excel is proposed.

1.1. Objectives

Since the following thesis is made in cooperation with the industry – Retech company, the goals standing behind it are correlated mainly with the needs of this company. As it is mentioned in an introduction above, production of *THSC* carries nowadays relatively high level of reliance on subcontractors because of the internally finned pipes. Rareness of this type of pipes influences the cost of the condenser and finally the cost of the *THSC* itself. Unfortunately, *THSC* on its own is only a subpart of another Retech product on sale. That is why the company cannot fully resign from this product but looks for improvements of their *THSC* design. Here, one can find the first general aim of this work – to find the way or tool helping in the process of improvement. Since the most important thing is the quality of the released air, it has to be defined which input parameters influence the performance of the condenser and, more important, what kind of values should be identified for verification of the right operation of *THSC*. The answer for the latter is the humidity, temperature and carried smells (or to be strict the lack of last in the outlet air). That is why output parameters like temperature and relative humidity of the released air have to be anticipated with respect to given inputs. Moreover, the user has to be informed about the appearance of a mixture of air/water mist at the outlet of *THSC,* which disqualifies its work as a condenser [4]. The model aims to reflect the *THSC* performance with actual design and when reducing the surface area of heat exchange – testing another prototype. Thanks to that, not only extreme working conditions can be tested but also customized versions of *THSC*. Since Retech company is not interested in either buying the license for CFD software, which is costly, or ordering consecutive researches from outsourcing companies, a mathematical model seems to be a better solution. This simplified *THSC* model is designed to generate a quick response for a wide range of inputs. The other reason standing behind the project is the availability of the Microsoft Office. Since mass and energy balances made to the entire device are insufficient to reflect the performance of *THSC* it is decided CFD model of the single pipe is performed. Having the results, it is possible to answer the question whether this model is necessary at all or tabular efficiencies for externally finned pipes can be used. Results of the CFD simulations for the single pipe are used as a base for a development of the single pipe thermal response model combined with global mass and energy balances in final *THSC* model. After this integration, major step is to validate the predictions of the *THSC* model against data from measurements [10]. It should be emphasized that the CFD model for *THSC* would give more accurate results than the present mathematical model. Nevertheless, the advanced numerical model is not fast enough and what is more important generates costs that can be avoided by manufacturer. That is why for the preliminary analyzes Retech company is willing to receive the mathematical model providing quick and rough analysis of the *THSC* work without demand for high computational power and huge investment. According to that, the main goal of the thesis is to answer the question whether a relatively simple model which can be used independently by Retech company is able to reflect the performance of *THSC*.

2. Top Hood Steam Condenser



1. Air inflow (blue arrow)	8. Left air collecting chamber	15 – side areas A_{side},(red plane)
2. Air filters	9. Right air collecting chamber	16 – top area A_{top}, (red plane)
3. Fan	10. Mixing chamber	17 – backside area A_{back} (red planes)
4. Distribution chamber	11. Air outflow (orange arrow)	18 – side walls in 6 (white)
5. Left condensation section	12. Condensate (navy arrow)	19 – side walls in 5 (white)
6. Right condensation section	13. Steam passage	20 – top wall in 6 (white)
7. Steam inflow (red arrow)	14. Vents	21 – top wall in 5 (white)

Figure 1 Top-side, top and side view of THSC

When analyzing the construction of *THSC*, two parts should be distinguished: cold (blue) and hot (red) in Figure 1. The former includes the air inlet (1) with filters (2) and a fan (3). The hot part consists of distribution chamber (4) and two condensation sections: the left (5) and right (6) where each consists of 24 pipes. Below the left condensation chamber, there are two steam channels (7). In addition, there can be also distinguished two heated air collecting chambers on the sides of condensation sections respectively (8-9) and the mixing chamber (10) located above the distribution chamber and condensation sections. Last two locations are air outflow (11) and condensate runoff (12). Starting from the cold part of the condenser the atmospheric air is sucked in (1) and flows through the filters (2) and a fan (3) to the distribution chamber (4) where it spreads between 48 pipes divided equally on the right (6) and left (5) sections of *THSC*. The air absorbs the heat from the steam flowing perpendicularly between the pipes. The warmed air leaving the tubes comes to the collecting chambers behind the appropriate sections and passes over them to the mixing chamber (10) from where it is directly released through the air outflow (12) to the surroundings of the device. The steam enters the *THSC* via steam channels (7) and is divided between two condensation sections. To the left section (5) steam comes directly while to the right one (6) it goes through the narrow steam passage (13). After the condensation process, the appearing condensate flows down to the oven through the condensate runoff (12). If some of the steam does not condensate passing through all 8 columns of the pipes then it is sucked by the fan through the small vents (14) installed on the wall that separates cold and hot part of *THSC*. That amount of steam then is mixed with air from the inflow.

3. Model

CFD model of a single pipe is prepared to estimate the dependence of the condensation on various working condition. First of all, it is checked that the 0.95 mm thick pipe (1) is made of steel and has 285.6 mm of length (2). Along the pipe of 28 mm internal diameter (3), there are 12 fins mounted evenly around the circumference. Each of the 1mm thick fin (4) has 6.25 mm of length (5). All the top and bottom curvatures of the fins are neglected and fins are assumed to be square. From the heat transfer point of view, the change is negligible. Numerically, for the mesh creation and Fluent calculation, such an assumption drastically improves the feasibility of the solution [3]. Although the geometry of the pipe is symmetrical, the air flows through the pipes of *THSC* in a non-symmetrical way under different angles. Since there is no certain information how the air flow rate will deflect inside the pipe, at this stage of model preparation the only way is to take the geometry of the whole pipe instead of a repeatable pipe fragment. To ensure good quality of the results it is decided to build a fully structural hexahedral mesh through the pipe [3] that consists of not more than half of a million elements (512,000 elements) since academic campus version of Ansys should be used. To eliminate doubts whether the solution depends on the number of elements and keep the limitation, it is decided to generate mesh consisting of 2 million elements which is validated and assumed as a benchmark for smaller ones. Final mesh is chosen as one with best quality of the results from all 13 tested and satisfactory enough comparing to benchmark.

3.1. Analytical model of the pipe heat transfer

Having the model ready, it is decided to take into consideration five parameters which may influence the pipe performance, which are: *T* – the temperature of air (T1 – 20°C to T6 – 40°C), *H* – relative humidity of air (H1 – 40% to H6 – 100%), *A* – the angle at which the air inflow the pipe (A1 – 10° to A6 – 80°), *M* – mass of the air flowing through the pipe (M1 –equal to 0.003958 kg/s up to M5 – equal to 0.007917 kg/s) and finally *S* – temperature of the steam (S4 – 100°C up to S1 – 90°C). Since the time and computational power for each simulation are limited matrix of solution is built on the basis of representative cases with the widest possible distribution of the parameters within chosen variants. It is decided to choose the base variant with 20 more modified versions. Each variant is named on the basis of the parameter and the number representing the value e.g. T3H3A4M3S4 – stands for the basic model assumed according to the measurements [4] as the most representative one with average or most probable values: T3(30°C), H3(60%), A4(45°), M3(0.003958 kg/s), S4(100°C) [10].

Table 1 Power output from Fluent simulation for all cases

Variant name	Q_{CFD} , W	Variant name	Q_{CFD} , W	Variant name	Q_{CFD} , W
T1H3A4M3S4	167.08	T3H1A4M3S4	146.36	T3H3A1M3S4	136.93
T2H3A4M3S4	156.94	T3H2A4M3S4	146.60	T3H3A3M3S4	140.36
T3H3A4M3S4	146.84	T3H3A4M3S4	146.84	T3H3A4M3S4	146.84
T5H3A4M3S4	136.79	T3H5A4M3S4	147.32	T3H3A5M3S4	152.62
T6H3A4M3S4	126.78	T3H6A4M3S4	147.80	T3H3A6M3S4	173.45
T3H3A4M1S4	81.25	T3H3A4M3S1	125.86	T6H6A5M1S1	63.05
T3H3A4M3S4	146.84	T3H3A4M3S2	134.25	T1H1A2M5S4	266.24
T3H3A4M4S4	200.54	T3H3A4M3S4	146.84	T4H4A6M2S3	129.92
T3H3A4M5S4	247.48				

3.2. Fin efficiency analysis

On the basis of the results from single pipe model, the efficiencies of the internal fins are calculated for a couple of cases. Comparison to tabular values shows the efficiencies differ and what is more for the internally installed fins values (82-88%) are always 5-6% lower than for externally installed ones (88-94% respectively). Combining this fact with the strong dependency of the pipe power on inflow angle indicates that the model relying solely on fin efficiency and velocity of the air is unreliable. The already existing formulas are not suitable for tubes internally finned.

3.3. THSC model development

To compute the empirical equation for model all the cases (Table 1) are divided into two groups. The first one gathers most of the cases with a representative range of change within parameters. The second group stand for validation of the final equation and consists of two opposite extreme conditions for heat exchange in the pipe (T6H6A5M1S1, T1H1A2M5S4) while the last one is randomly chosen with all values not set previously (T4H4A6M2S3). Those three cases do not take part in equation creating process. At first sight, the biggest increase in power could be noticed when changing the mass flow rate of the air. Nevertheless, such a hypothesis is verified testing a few variants of empirical equation. Finally, a variant based on mass flow rate correlation (1) with air humidity taken into account is chosen as model equally good in the whole range of observed changes (lowest maximum error at the level of 1.5%).

$$Q_{pipe} = (-1190589 \cdot M^2 + 39693.5039 \cdot M + 7.6052) \cdot \varepsilon_T \cdot \varepsilon_A \cdot \varepsilon_S \cdot \varepsilon_H \quad (1)$$

$$\varepsilon_T = \frac{-2.0148016 \cdot (T - 273) + 207.32898}{146.84}$$

$$\varepsilon_A = \frac{0.0077559123 \cdot A^2 - 0.1938093674 \cdot A + 138.5566319398}{146.84}$$

$$\varepsilon_S = \frac{2.098028 \cdot (S - 273) - 62.964247}{146.84}$$

$$\varepsilon_H = \frac{0.02390924 \cdot H + 145.40501207}{146.84}$$

$\varepsilon_T, \varepsilon_A, \varepsilon_S, \varepsilon_H$ – correction factors with respect to: M - mass flow rate of the air flowing through the I pipe, T - air temperature, A - angle between direction of inflowing air and central axis of pipe, S - steam temperature, H - relative humidity of air

There are two more phenomena verified as negligible, namely heat loss (2) through the external walls (15-17 in Figure 1) and heat gain (3) from the condensation chamber's walls (18-21 in Figure 1).

Q_{lost} equal to 80 W and Q_{gain} equal to 285 W are put into the code as constants.

$$Q_{lost} = q_{air} \cdot \left(2 \cdot A_{side} + A_{top}\right) + q_{steam} \cdot A_{back} \tag{2}$$

$$Q_{gain} = \dot{m}_{cond} \cdot h_{fg} \tag{3}$$

q_{air}, q_{steam}– heat flux for air and steam with radiation taken into account W/m^2, $\dot{m}_{cond}$ – mass flow rate of steam condensated on the walls of condensation chambers in *THSC* kg/s

The angles and shares of total air mass flow rate between the pipes (4 in Figure 1) are implemented in *THSC* model from full CFD model [10] as the default setup. However, the user is given the option to modify either particular angles or shares of inflowing mass flow rate. Unequal distribution of steam (13 in Figure 1) between condensation chambers (5 and 6 in Figure 1) is controlled by the equation (4) according to measurements [10] with the assumption for values of D_f higher than 0.5 the model sets equal distribution of steam.

$$D_f = 7.5508 \cdot \dot{m}_{steam} + 0.3848 \tag{4}$$

D_f – the share of steam mass flow rate directed to the right section of THSC, $\dot{m}_{steam}$ – total steam mass flow rate kg/s

Direct use of single pipe model in *THSC* model would lead to the maximum condensation of steam which does not coincide with the results of measurements. For smaller steam flow rate there are areas where there is no contact between steam and pipe – which reduces the power of the pipe. That is why the correction factors (*Eps*) are introduced to control how much of the steam is condensed on each pipe depending on steam flow rate left in the condenser. Therefore, from fully developed CFD model of *THSC* which is validated against measurements [10] the power values for all pipes are compared with the ones from *THSC* model. Calibrated *THSC* model further on is validated against measurements [10]. In consequence, the limitation works only when the amount of steam is relatively small, close or below the maximum available amount of steam condensed on the pipe. When the flow rate is much bigger, calculated limits for *Eps* are above physical limits for a single pipe and do not activate.

The general procedure of calculation in *THSC* model starts from steam mass flow rate division between the left and right condensation chamber according to internal formula (4) or equally and decreased by the amount of steam condensed on the walls (3). For the air mass flow rate entering the distribution chamber (4 in Figure 1) of *THSC* the saturation pressure is read. Then the degree of humidity (5) and specific enthalpy (6) are calculated. Furthermore, the calculation for pipes in left (and then right) section of condensation chamber is repeated 24 times. For given input parameters the power of the pipe is calculated (1) as well as the corresponding amount of condensed steam (7). The same value is validated against *Eps* and modified if needed. Next, the specific enthalpy (8) of warmed air is calculated from energy balance. Since there is no mixing and degree of humidity (5) does not change the temperature, and relative humidity of air at the outflow are calculated using reformulated equations (6) and (5) respectively. In the end, the amount of steam left for calculation of next pipe is decreased according to equation (9). After all 24 calculations for pipes from left (then right) condensation chamber are done there is a second subpart of *THSC* to be calculated. Namely, all 24 air mass flow rates get to the collecting chamber and are mixed. Therefore, mass (10) and energy (11) balances are used to calculate the degree of humidity and specific enthalpy of air leaving the left (then right) collecting chamber. Similarly, as in the case of condensation chamber here the temperature and relative humidity of mixed air flow rates at the outflow of collecting chamber are calculated with reformulated (6) and (5) respectively. In this place, it is checked for the first time whether the mist appears in the air released from left collecting chamber (then right one). To verify it the result of the degree of humidity (10) is compared with the maximum degree of humidity (12) for these conditions. If (10) is higher than (12) the temperature is found iteratively comparing (11) with equation (13) which takes into account mist in the air. Finally, having the results from both sections the equations for outflow of *THSC* are calculated at the end of the code. One more time the degree of humidity (14) and specific enthalpy of air at outlet (15) are calculated using mass and energy balances. For this subpart of the *THSC* also the maximum degree of humidity like in (12) is calculated and compared with (14). In case of mist appearance the value of specific enthalpy (15) is recalculated and which is more important the user is informed about critical failure of *THSC* performance.

$$X_{in} = \frac{\frac{H_i}{100} \cdot p_{sat,in,i}}{p_o - \frac{H_i}{100} \cdot p_{sat,in,i}} \cdot \frac{M_{H2O}}{M_{da}} \tag{5}$$

$$h_{wa,in,i} = 1.005 \cdot (\mathrm{T} - 273.15) + X_{in} \cdot (1.88 \cdot (\mathrm{T} - 273.15) + 2501) \tag{6}$$

$$\dot{m}_{cond,i} = \frac{Q_{pipe,i}}{h_{fg}} \tag{7}$$

$$h_{wa,out,i} = \frac{h_{wa,in,i} \cdot \dot{m}_{air,i} + h_{fg} \cdot \dot{m}_{cond,i}}{\dot{m}_{air,i}} \tag{8}$$

$$\dot{m}_{steam\_left,i+1} = \dot{m}_{steam\_left,i} - \dot{m}_{cond,i} \tag{9}$$

$$X_{out,L} = \frac{\sum_{24}^{1} \dot{m}_{air,i} \cdot X_{in} + \left(\dot{m}_{steam\_left,1} - \sum_{24}^{1} \dot{m}_{cond,i}\right)}{\sum_{24}^{1} \dot{m}_{air,i}} \tag{10}$$

$$h_{wa,out,L} = \frac{\sum_{24}^{1} h_{wa,out,i} \cdot \dot{m}_{air,i} + \left(\dot{m}_{steam\_left,1} - \sum_{24}^{1} \dot{m}_{cond,i}\right) \cdot h_S}{\sum_{24}^{1} \dot{m}_{air,i}} \tag{11}$$

$$X_{max} = \frac{p_{sat,out,L}}{p_o - p_{sat,out,L}} \cdot \frac{M_{H2O}}{M_{da}} \tag{12}$$

$$h_{wa,out,L} = 1.005 \cdot \left(T_{out,L} - 273.15\right) + X_{max} \cdot \left(1.88 \cdot \left(T_{out,L} - 273.15\right) + 2501\right) + (X_{out,L} - X_{max}) \cdot 4.19 \cdot \left(T_{out,L} - 273.15\right) \tag{13}$$

$$X_{out} = \frac{\sum_{48}^{1} \dot{m}_{air,i} \cdot X_{in} + \left(\dot{m}_{steam} - \dot{m}_{cond} - \sum_{48}^{1} \dot{m}_{cond,i}\right)}{\sum_{48}^{1} \dot{m}_{air,i}} \tag{14}$$

$$h_{wa,out} = \frac{\sum_{48}^{1} h_{wa,out,i} \cdot \dot{m}_{air,i} + \left(\dot{m}_{steam} - \dot{m}_{cond} - \sum_{48}^{1} \dot{m}_{cond,i}\right) \cdot h_S - Q_{lost} + Q_{gain}}{\sum_{24}^{1} \dot{m}_{air,i}} \tag{15}$$

$p_{sat,in,i}, p_{sat,out,L}$ – saturation pressure for $T, T_{out,L}$ hPa, M_{H2O}, M_{da} – molar mass of water and dry air $kg_{H2O}/kmol_{H2O}$, $kg_{da}/kmol_{da}$, p_o – air pressure hPa, $\dot{m}_{cond,i}$ – mass flow rate of steam condensed on the i pipe in *THSC* kg/s, i – number of calculated pipe 1-24, $\dot{m}_{air,i}$ – mass flow rate of the air flowing through i pipe in *THSC* kg/s, $\dot{m}_{steam\_left,i}$ – mass flow rate of steam left to be condensed on the i pipe in *THSC* kg/s, h_S- enthalpy of steam for $T_{steam}, (S)$ kJ/kg, $T_{out,L}$ – temperature of mixed air flow rates at the outflow of collecting chamber K

Validation of the model is provided for two *THSC* working conditions. Firstly the default work is examined (all pipes are active and the steam distribution is controlled internally by equation (4) dependent on its flow rate) [10]. Discrepancies between measured outputs and generated in the *THSC* model are at the satisfying level. What is more important they show the right trend of changes with input values. It can be observed e.g. for the condensate in Figure 2 where both lines correspond very well to each other with the exception of four cases (11,12,13,15) only which can be related to defective measurements. Secondly, according to predictions it is tested how the model behaves when some of the pipes are cut off to enlarge steam passage (prototype with an improved construction of *THSC* [10]). Switching off the 1st and 8th tier in both sections it is tested whether the *THSC* model can visualize the behaviour of *THSC* with changed geometry influencing the steam distribution. Looking at condensate when 1st and 8th rows of pipes are turned off it can be observed in Figure 2 that values from measurements oscillate close to the ones from *THSC* model except cases 9 and 10 (red box). For those

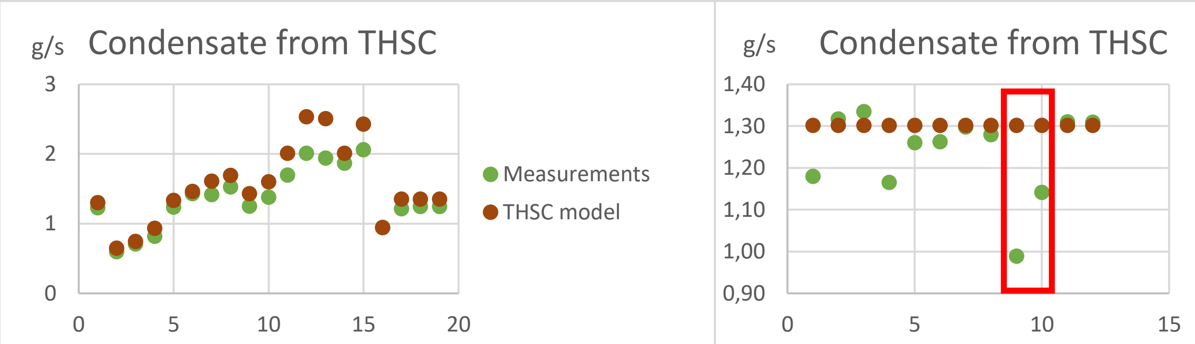


Figure 2 Condensate results from THSC model vs measurements (validation left view, prototype right view)

measurements, there is a problem with the amount of condensate. There is no other explanation for much smaller condensation as the operating conditions are similar to case 5 and 6. Amount of condensate calculated in *THSC* model for such a small steam flow rate is insensitive to parameters other than order of switched off/on rows of pipes or steam flow rate itself. According to all analyses it is observed that the model is not perfectly corresponding to every measurement [10]. Nevertheless, *THSC* model can be assumed as an adequate tool to observe trends of change. Imperfections depending on simplification are small enough to be accepted and worth time saved in comparison to the full model. But of course, it has to be highlighted that such a model cannot be used as one and only tool for making final decisions. Any change based on *THSC* model analysis before carrying it out has to be verified in real scale prototype or at least fully developed CFD model.

4. Analysis of THSC work

The *THSC* is aimed to condensate steam and eliminate the risk of a mixture of air/water mist appearance at the outlet. Therefore, the analysis takes into account worsening conditions favorable water droplets creation at the outlet. At the inlet default values are kept for air mass flow rate (0.208019 kg/s), pressure (987.2 hPa), steam mass flow rate (0.00136 kg/s) and temperature of steam (373.15 K). At the beginning it is checked that both: increase of RH of incoming air up to 100% and increase of temperature up to 307 K does not change the amount of condensate keeping the condensation efficiency at the level of about 99%. When additionaly the steam flow rate is increased up to 0.00408 kg/s the number of condensate increases. When the same amount of steam is even higher *THSC* works with maximum possible power since there is enough steam inside it to be sure pipes are all the time surrounded by steam. Thenceforth, the amount of condensate does not change and each additional portion of steam results in both increased humidity and decreased condensation efficiency. Even though, the bigger amount of steam would be most probably limited in reality much earlier by fluid resistance, according to assumed values it is decided to check for what amount of steam the mixture of air/water mist appears. Table 2 presents case when this phenomenon takes place (steam mass flow rate equal to 0.12784 kg/s).

Table 2 THSC work analysis - outflow results for change of steam flow rate (mixture of air/water mist appearance)

Right Section			Total outflow from THSC			Left Section		
Condensate	T_{air}	RH	Condensate	T_{air}	RH	Condensate	T_{air}	RH
kg/s	K	%	kg/s	K	%	kg/s	K	%
1.65E-03	354.7	100.00	3.53E-03	354.5	100.00	1.75E-03	354.3	100.00

Feedback given by above analysis proves the design of *THSC* can provide proper work and fulfill its task under much worse condition than commonly occurred. Therefore, it raises the question if some of the pipes were cut off would the condenser still satisfy the customer demands. One of the approaches is examined for default values (except RH=100% and T_{air}=307 K) with the same, unchanged amount of inlet air which is equally redistributed between the rest off working pipes

Table 3 THSC work analysis - outflow results for change of pipes in right section with internal formula for distribution of steam

Switched off rows in right section	Right Section			Total outflow from THSC			Left Section		
	Condensate	T_{air}	RH	Condensate	T_{air}	RH	Condensate	T_{air}	RH
	kg/s	K	%	kg/s	K	%	kg/s	K	%
-	4.85E-04	317.0	58.00	1.35E-03	320.4	49.00	7.43E-04	321.8	46.00
1	4.84E-04	318.0	56.00	1.35E-03	320.4	49.00	7.43E-04	320.6	49.00
1,2	4.83E-04	319.1	53.00	1.35E-03	320.3	49.00	7.43E-04	319.7	51.00
1,2,3	4.80E-04	320.6	49.00	1.35E-03	320.3	49.00	7.43E-04	318.7	54.00
1,2,3,4	4.74E-04	322.9	44.00	1.34E-03	320.2	50.00	7.43E-04	317.8	56.00
1,2,3,4,5	4.68E-04	326.7	36.00	1.34E-03	320.2	50.00	7.43E-04	316.9	59.00
1,2,3,4,5,6	4.47E-04	333.3	27.00	1.32E-03	320.0	50.00	7.43E-04	316.1	61.00
1,2,3,4,5,6,7	2.84E-04	338.6	26.00	1.15E-03	318.4	56.00	7.43E-04	315.3	64.00
All pipes	0.00E+00	373.0	100.00	8.69E-04	309.8	91.00	7.43E-04	314.5	67.00

It is assumed only rows of pipes from the right section are switched off. Steam is distributed according to internal formula (4). On the basis of THSC model for assumed input data right section can be fully switched off and the left one is enough to prevent the mixture of air/water mist at the outflow. Condensation efficiency drops below 99% when the 4th row of pipes is switched off and for only left section working the same factor is equal to 63.9%. Results are stored in Table 3.

5. Conclusions

One of the most important conclusions of the thesis concerns creation of the single pipe model. Results from model and tables shows internally installed fins are 5-6% less efficient than in theory. It proves there is no reliable correlation for internally installed fins and reasonable is to model the work of single pipe. Even though there are some discrepancies comparing results from the model to the measured ones [10], model let the user observe right trends of changes. *THSC* is most probably overscaled since even much worse operating conditions does not result in a mixture of air/water mist appearance at the outflow which is disqualifying factor. Namely, for the inlet air of 100% relative humidity and 307 K, the RH at the outflow increase by only 3 percentage point and the condensation efficiency does not drop below 99%. Increasing additionally the steam mass flow rate the amount of humidity does not exceed the saturation point unless the mass flow rate is 94 times greater (0.12784 kg/s) than default one (0.00136 kg/s). Such a huge flow rate most probably would not flow through a device without any change in fluid resistance. Three different approaches are verified turning off the rows of pipes in both sections. One approach proves *THSC* can work with only one (left) section turned on. With default operating conditions the mixture of air/water mist does not appear yet at the outflow and the condensation efficiency drops down to the level of 63.9%. The condensation efficiency is kept at the level of 99% until the 4th row of pipes in right section is switched off. This analysis confirms, with no consequence user can reject first 3 rows of pipes. Since *THSC* is only a subpart of large scale cooking oven, it is highly recommended to consider a few customized versions of *THSC* depending on designed operating conditions. Such an action may decrease the investment cost and does not hamper the fitting of *THSC* as subpart. One of the main conclusions of the thesis is that final version of *THSC* model satisfies the Retech company needs to have software for private use free of paid license and straightforward for users without specific technical background. Work with *THSC* model can lead to better device performance and reduced expenditures thanks to quickly forecasted results of changes.

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