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# Optimization of a Cooling-Condensation Tower

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Cooling condensation towers are used in many different industrial applications such as in power and incineration plants as well as in the chemical and paper industry in order to recover heat from a flue gas stream and therefore significantly increase the energy efficiency of the plant. One widespread approach to achieve this goal is gas cooling and condensation through direct contact between liquid and gas streams in a packed bed.
The dimension of such cooling units, and thereby the investment and operating costs are directly dependent on the selected mass transfer components. Key performance parameters of these are pressure drop, hydraulic capacity as well as heat and mass transfer efficiency.
Mass transfer equipment manufacturers have the possibility to optimize the size of these cooling towers through selection of best suitable mass and heat transfer components. In this work, a design case of a cooling-condensing tower equipped with different random and structured packings made out of different materials will be presented. For this purpose, maximum pressure drop and temperatures of the inlet streams in the condensing stage will be specified and the packing type varied. The derived investment costs will be evaluated.

## 1. Introduction

Direct contact heat transfer between counter currently flowing gas and liquid streams is used in many different industrial applications. In the pump-around sections of refinery columns or ethylene quenchers, hydrocarbon gas and liquid phases at process conditions different than atmospheric are involved. In many other cases, on the contrary, such as in power and incineration plants as well as in the chemical and paper industry, heat is transferred/recovered from a flue gas stream to a water stream at nearby atmospheric conditions. The heat recovery achieved significantly increases the energy efficiency of the plant. In wood fired boiler houses, the efficiency increase is estimated to reach 5-10% (Veidenbergs et al., 2010).
Packed bed columns are preferentially used as direct contact devices in these applications because of their simple configuration (reduced capital investment costs) and ease of operation (less sensitivity to corrosion and fouling problems leading to reduced operation and maintenance costs compared to classical heat exchangers). The dimension of such cooling stages and the investment and operating costs are directly dependent on the selected mass transfer components, i.e. packings.
For the design of a gas cooling stage, the simultaneous heat and mass transfer must be taken into consideration in combination with the packing hydraulics. Different gas cooling types are distinguished depending on the nature of the heat transferred:

- Gas cooling without significant mass transfer: heat is mostly transferred as sensible heat due to a temperature gradient.
- Gas cooling with evaporation: heat is removed from the gas by evaporation of part of the liquid.
- Gas cooling with condensation: heat is extracted from the gas stream by liquid condensation and gas cooling simultaneously.

In this work, a design case of a cooling-condensing tower equipped with different random and structured packings made out of different materials with a similar specific surface area around 100 $m^2/m^3$ will be presented to show the impact of the selected packing on tower dimensions and investment costs of the heat recovery unit.

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## 2. Design case – Cooling-Condensing tower

A typical cooling-condensing design case has been selected from a real industrial application to estimate the column dimensions with different packings commercially available at RVT Process Equipment GmbH and evaluate the most advantageous of them regarding investment costs. Figure 1 shows a principle sketch of this design case.

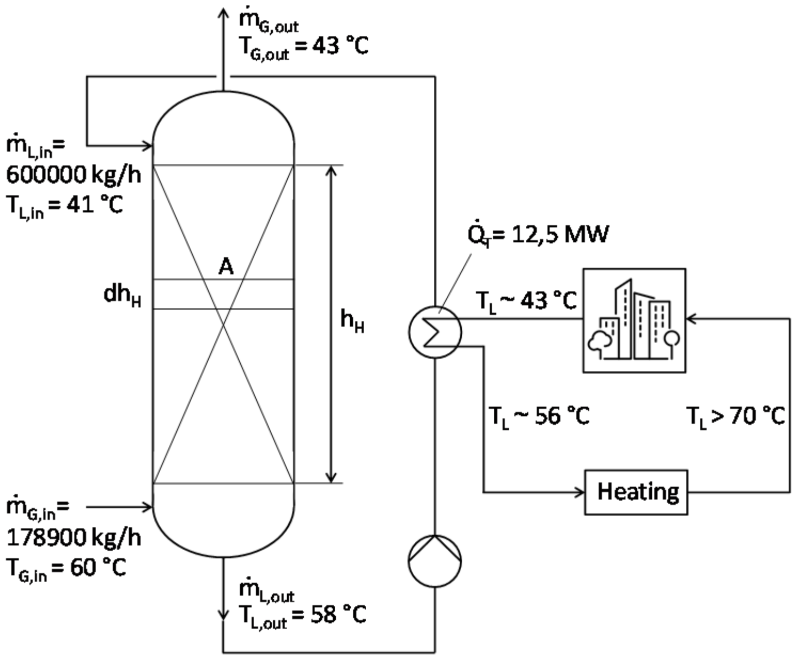


*Figure 1: Principle sketch of the condensing-cooling process selected for the design case study*

A (flue) gas stream, $\dot{m}_{G,in}$, of 178900 kg/h saturated with water at 60 °C ($T_{G,in}$) is available from which 12,5 MW ($\dot{Q}_T$) energy needs to be transferred to a liquid stream so that it can be used elsewhere as a heating source. To achieve this, a packed tower is used, where the gas and liquid streams are fed at the bottom and top of the bed respectively and flow counter currently to each other. Heat is transferred from the gas to the liquid stream by direct contact within the packed bed through condensation (latent heat) and cooling (sensible heat). According to the amount of energy to be removed, gas leaves the packed bed at a temperature of $T_{G,out}$= 43 °C. Therefore, the liquid stream inlet temperature required to cool down the gas must be lower. A heat balance within the cooling stage shows that the higher the liquid stream inlet temperature, the higher the mass flow necessary to achieve the targeted cooling task. Moreover, in order to be able to use the recovered heat elsewhere, the liquid stream outlet temperature must be as high as possible. Therefore a liquid inlet temperature of $T_{L,in}$= 41 °C was chosen, i.e. 2 K temperature gradient between gas and liquid at the top of the packed bed, for the case study. Finally a liquid water flow of $\dot{m}_{L,in}$= 600000 kg/h, was selected so that the achieved liquid outlet temperature calculated from the heat balance is around $T_{L,out}$= 58 °C, this is, approximately 2 K lower than the gas inlet temperature. The mean logarithmic temperature difference between gas and liquid along the packed bed is therefore approximately 2K.
Outside the column, this liquid stream is typically fed to a heat exchanger where a second cooler liquid stream from an independent circuit is introduced which heats up to a similar temperature level whereas the originally hotter liquid stream cools down and is fed back to the cooling-condensing tower circuit. The independent water circuit stream leaving the heat exchanger at a temperature around $T_L$= 56 °C can be further heated up by water steam or a heat pump system to increase the temperature level to above 70 °C, and be used as heating source for different purposes such as in a district heating network.

### 2.1 Heat transfer in cooling-condensing towers

When gas cooling with condensation takes place, the total heat removed ($\dot{Q}_T$) is the sum of the sensible heat ($\dot{Q}_S$) and the condensation heat or latent heat ($\dot{Q}_{Cond}$). As Eq(1) shows, the sensible heat is calculated as the product of the total gas mass flow, $\dot{m}_G$, the gas specific heat capacity, $c_{P,G}$, and the difference between gas inlet and outlet temperature. The latent heat is calculated from the product of the condensed water molar flow, $\dot{n}_{Cond}$, and its molar condensation enthalpy, $\Delta H_{LV}$.

$$\dot{Q}_T = \dot{Q}_S + \dot{Q}_{Cond} = \dot{m}_G\, c_{P,G}\, \left(T_{G,in} - T_{G,out}\right) + \dot{n}_{Cond}\, \Delta H_{LV} \tag{1}$$

According to the two-film theory, the heat flow can be expressed as the product of the temperature difference between the gas and liquid streams, the overall heat transfer coefficient, $\alpha_{OG}$, the effective heat transfer area per unit volume, $a_{eff}$, and the differential volume element for the heat transfer, $dV_H(A\, dh_H)$:

$$dQ = \alpha_{OG} a_{eff} (T_G - T_L) dV_H \tag{2}$$

For a cooling condensing process, following expression for the overall heat transfer coefficient results:

$$\alpha_{OG} = \frac{1}{\dfrac{1}{\alpha_L} + \dfrac{1}{E_T \alpha_G} \dfrac{\dot{Q}_S}{\dot{Q}_T}} \tag{3}$$

where $\alpha_L$ and $\alpha_G$ are the liquid and gas phase heat transfer coefficients and $E_T$ is the Ackermann correction, defined as:

$$E_T = \frac{\emptyset_T}{1 - e^{(-\emptyset_T)}} \quad , \quad \emptyset_T = \frac{\dot{n}_{Cond}\, \tilde{c}_{P,G}}{a_{eff}\, V_H\, \alpha_G} \tag{4}$$

where $\tilde{c}_{P,G}$ is the gas molar heat capacity and $V_H$ the total packing volume for the heat transfer process. The Ackermann correction considers the additional heat transfer which takes place by condensation. For condensation, the mass flow direction of the condensed stream and the heat flow direction are the same (gas to liquid), and the molar flow takes a positive value, so that the Ackermann correction takes a value higher than one.

The height required to realize a defined heat transfer process, $h_H$, can be calculated from the HTU-NTU concept by Chilton and Colburn (Chilton et al., 1935) and the analogy between mass and heat transfer (Colburn 1939):

$$h_H = \frac{V_H}{A} = NTU_{H,OG}\, HTU_{H,OG} \tag{5}$$

The number of heat transfer units, $NTU_{H,OG}$, is calculated from the relationship between the temperature change in the gas stream to the mean logarithmic temperature difference between gas and liquid along the packed bed:

$$NTU_{H,OG} = -\int_{T_{G,in}}^{T_{G,out}} \frac{dT_G}{T_G - T_L} = \frac{T_{G,in} - T_{G,out}}{\Delta T_{lm}} = \frac{T_{G,in} - T_{G,out}}{\dfrac{\left(T_{G,in} - T_{L,out}\right) - \left(T_{G,out} - T_{L,in}\right)}{ln \dfrac{\left(T_{G,in} - T_{L,out}\right)}{\left(T_{G,out} - T_{L,in}\right)}}} \tag{6}$$

The height of a heat transfer unit referred to the gas side, $HTU_{H,OG}$, can be calculated from the gas mass flow and heat capacity, the overall heat transfer coefficient, the effective packing area and the column cross section area, $A$, by following equation:

$$HTU_{H,OG} = \frac{\dot{m}_G\, c_{P,G}}{\alpha_{OG}\, a_{eff}\, A} \tag{7}$$

Therefore, in order to calculate the packed bed height necessary for a cooling-condensing case design, the overall heat transfer coefficient needs to be precisely estimated. As Eq(3) shows, the overall heat transfer coefficient in a cooling-condensing stage can be calculated if the heat transfer coefficients for the liquid and gas phase are known. Well proven models such as Onda, Schultes or Mersmann can be used for the estimation for the gas/liquid mass transfer coefficients from which, by the analogy between mass and heat transfer, and using the Lewis relationship, the heat transfer coefficients can be estimated. For the air-water system, for which a Lewis number of approximately one applies, this relationship results:

$$\beta_{L/G} = \frac{\alpha_{L/G}}{c_{P,L/G}\, \rho_{L/G}} \tag{8}$$

For the cooling-condensing case studied in this work, the number of transfer units is only dependent on the gas and liquid inlet and outlet temperatures and therefore independent of the specific packing type used. Packing height necessary will only depend on the calculated $HTU_{OG}$ and therefore on the specific packing type selected. From Eq(7) results that $HTU_{OG}$ will change with the packing type chosen due to differences arising from the estimated overall heat transfer coefficient, packing effective area and tower diameter (section area).

### 2.2 Packings studied

Table 1 and Figure 2 show the types and main geometrical characteristics of the studied packings. Two plastic random packings, one metal structured packing and an hybrid plastic packing have been considered. All four packings are manufactured and commercialized by RVT Process Equipment GmbH in Steinwiesen (Germany). Since generally speaking mass and heat transfer is proportional to the geometrical (effective) packing area, the four packings chosen have been selected so that the specific surface area is in a similar absolute range around 100 $m^2/m^3$ (between 75 and 125 $m^2/m^3$). As a result, a fair comparative analysis of column size differences and investment costs can be performed. Furthermore, all four selected packings can be considered as low pressure drop-high capacity packings allowing performance up to high liquid and gas loads, typical in cooling-condensing processes. It is therefore that precisely these are the packing types and sizes that come into consideration for the design of such cooling-condensing towers in the industrial practice.

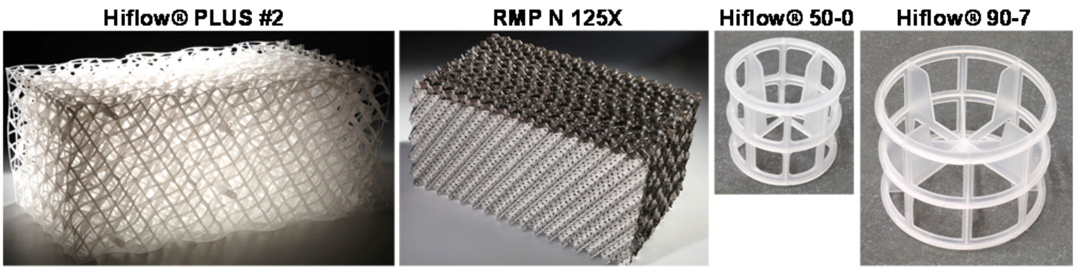


*Figure 2: Packings considered for the cooling-condensing case study*

*Table 1: Geometrical characteristics of the packings considered for the cooling-condensing case study*

| Packing type | $a_{geo}$ / $m^2/m^3$ | ε / - | Material |
|---|---|---|---|
| Hiflow® 50-0 | 110 | 0,947 | plastic (PP) |
| Hiflow® 90-7 | 76 | 0,970 | plastic (PP) |
| Hiflow® PLUS#2 | 100 | 0,950 | plastic (PP) |
| RMP N 125X | 125 | 0,987 | metal (AISI 304) |

As random packing, the industrially widely used Hiflow® ring with two different nominal sizes: 50 and 90 mm has been selected. Hiflow® 50-0 has a specific surface area of 110 $m^2/m^3$ and a void fraction around 95 %. The bigger sized Hiflow® 90-7 offers a lower specific surface area of 76 $m^2/m^3$ combined with a higher void fraction of 97 % allowing for lower pressure drop – higher capacity. Hiflow® PLUS#2 is an hybrid plastic packing consisting of an ordered structure based on the highly open Hiflow® 50-0 geometry, differing therefore substantially from conventional corrugated sheet packings. The highly open ordered structure enables a free exchange of vapour and liquid in the horizontal direction. Hydraulic and mass transfer characteristics of this grid-type packing have been presented elsewhere (Lehner et al. 2011) and show that significantly lower pressure drops and higher capacities combined with similar mass transfer efficiencies compared to random packings are achieved. The fourth considered packing: RMP N 125X, is a conventional structured metal packing with 125 $m^2/m^3$ specific area and 60° corrugation angle to the horizontal (X-type). It must be pointed out that for the selected air-water system and operating conditions, where gas inlet temperatures are around 60 °C and below, PP is the preferred material of construction due to lower manufacturing costs. In this case, however, the metal structured packing has been included in the study since its particular geometrical features: very high void fraction (ε = 0,987, highest among the selected packings) and steep X-type corrugation angle allow for very low specific pressure drop and very high capacity figures which make this packing worth to be considered as a packing candidate for the cooling-condensing tower design and total investment analysis. A disadvantage might arise from the higher investment costs for corrugated sheet packings in general.

### 2.3 Calculation procedure

Figure 3 shows the schematic sequence to calculate the total cooling-condensing tower investment costs.

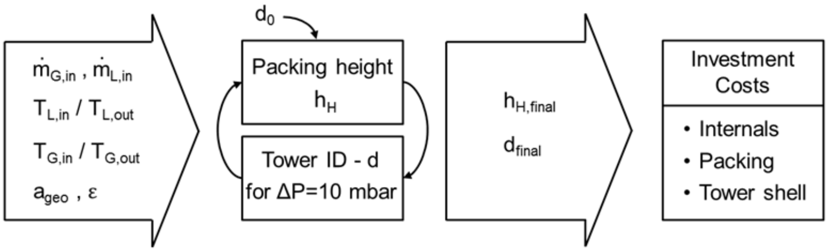


*Figure 3: Schematic calculation sequence of the total cooling-condensing tower investment costs*

For the known process conditions: gas and liquid inlet flows, gas and liquid inlet and outlet temperatures and physical properties of each selected packing type, an initial tower inside diameter is assumed and the overall heat transfer coefficient calculated. This allows for estimation of the packed bed height necessary using the HTU-NTU model as described in chapter 2.1. Using the hydraulic rating software Rapsody 3.10 (in-house product developed by RVT Process Equipment GmbH freely available at www.rvtpe.com), the total pressure drop resulting for the operating conditions in the estimated column size and bed length is calculated. As boundary condition the total pressure drop of the packing is set to exactly 10 mbar, so that there is no difference in the operating costs and just the investment costs are considered on a comparative basis. If the targeted value is not achieved, a new tower diameter is accordingly chosen and the calculation sequence re-started. Finally a specific tower diameter and packed bed length for each packing type is obtained.

From the specific tower dimensions obtained for each packing type, the investment costs for tower internals (support grid, hold-down grid, liquid distributor including feed pipe and droplet separator), packing and tower shell are estimated. Tower shell investment cost is based on fibreglass reinforced plastic (GRP) as material of construction, which is mostly used in this kind of applications whereas material considered for the tower internals is polypropylene (PP).

## 3. Results

Table 2 shows the results of the cooling-condensing stage design and investment costs for each of the four selected packings.

*Table 2: Results for cooling-condensing stage design and investment costs for the selected packings*

| Packing type | Heat transfer coeff.–$\alpha_{OG}$ / $W/m^2K$ | Tower ID $d_{final}$ / mm | Packing height–$h_H$ / m | Packing volume–$V_H$ / $m^3$ | Liquid load–$u_L$ / $m^3/m^2h$ | Gas Factor $F_G$ / $Pa^{0,5}$ | Relative Invest. costs / - |
|---|---|---|---|---|---|---|---|
| Hiflow® 50-0 | 555 | 5660 | 5,6 | 141 | 24,8 | 2,39 | 1,00 |
| Hiflow® 90-7 | 506 | 4820 | 7,3 | 133 | 34,2 | 3,29 | 0,89 |
| Hiflow® PLUS#2 | 625 | 4810 | 6,0 | 109 | 34,3 | 3,31 | 0,87 |
| RMP N 125X | 662 | 4470 | 6,1 | 96 | 39,8 | 3,83 | 1,13 |

The results show that the design which requires the largest tower diameter: 5660 mm, is the one with Hiflow® 50-0 whereas the smallest tower diameter design, 4470 mm, is achieved with a packed bed consisting of the metal structured packing RMP N 125X. This represents a 38 % reduction in the tower sectional area. Tower diameter with Hiflow® 90-7 and Hiflow® PLUS#2 are almost identical and only around 14 % larger than with RMP N 125X. This fulfils the expectation derived from the loading capacity of these four packing types.

Calculated overall heat transfer coefficient varies between 506 $W/m^2K$ for Hiflow 90-7 and the highest value of 662 $W/m^2K$ estimated for RMP N 125X. Hiflow® PLUS#2´s overall heat transfer coefficient is 625 $W/m^2K$, very close to the metal structured packing, whereas that for the Hiflow® 50-0 is 555 $W/m^2K$, closer to the value estimated for Hiflow® 90-7. These results can be explained on the background for the estimation of the overall heat transfer coefficient. As shown by Eq(3), the overall heat transfer coefficient for a specific packing type in a defined cooling-condensing case, only depends on the liquid side and gas side heat transfer coefficients. These are calculated from the individual mass transfer coefficients with the Lewis equation, Eq(8), as shown in section 2.1. As it is known, individual mass transfer coefficients predicted by the conventional models are directly dependent on the packing effective surface area. Therefore a lower mass/heat mass transfer coefficient should be expected for the packing with the lowest geometrical surface area, Hiflow® 90-7 as it is the case. For the same reason however, overall heat transfer coefficient expected for Hiflow® 50-0 should be larger, i.e., closer to Hiflow® PLUS#2 and RMP N 125X. The reason for this anomaly is that the models also predict an increase in the mass/heat transfer coefficient with increasing gas and liquid loads. As Table 2 shows, the considerably larger tower diameter with Hiflow® 50-0 results in significantly lower gas and liquid loads and therefore a lower overall heat transfer coefficient despite of its rather high specific surface area of 110 $m^2/m^3$.

Packing heights estimated also show significant differences between the four selected packings. Now, design with Hiflow® 50-0 delivers the lowest packed bed height: 5,6 m. On the contrary, the highest value: 7,3 m, is obtained for Hiflow® 90-7 whereas the height for Hiflow® PLUS#2 and RMP N 125X is almost identical: 6,0 and 6,1 m respectively. These figures can be explained from Eq(7) for the calculation of $HTU_{OG}$. According to Eq(5), the packing height is estimated from the $HTU_{OG}$ and $NTU_{OG}$ values. The number of transfer units is only dependent on the gas and liquid inlet and outlet temperatures and therefore for the present case study independent of the specific packing type used. Packing height is therefore determined by the $HTU_{OG}$ obtained for each packing type. From Eq(7) results that $HTU_{OG}$ will differ with the packing type selected from the different estimated overall heat transfer coefficients, packing effective area and tower diameter (tower cross section area) selected. For Hiflow® 50-0, the lowest necessary packing height results from the significantly larger tower diameter, which compensates for the lower heat transfer coefficient and effective area under lower gas and liquid loads. For Hiflow® 90-7 on the contrary, the significantly lower heat transfer coefficient together with the lower effective packing area result on a higher packed bed of 7,3 m despite of the smaller sectional area. Finally, for Hiflow® PLUS #2 and RMP N 125X all these three parameters nearly compensate so that the similar height of ca. 6,0 m is delivered.

Finally, last column in Table 2 shows the investment costs derived from the estimated cooling-condensing tower design with each packing type relative to the costs with Hiflow® 50-0. Highest investments are estimated for the metal structured packing RMP N 125X, despite the fact that the smallest tower diameter with a packed bed height of 6,1m, mainly due to the significantly higher packing costs compared to the other three considered plastic packings. Lowest investment costs are achieved with the hybrid structured packing Hiflow® PLUS#2 followed very close by Hiflow® 90-7, which deliver a ratio to Hiflow® 50-0 of 0,87 and 0,89 respectively. Costs with Hiflow® 50-0 and RMP N 125X are 15 % and 30 % higher respectively. Taking into consideration that Hiflow® PLUS#2 offers the additional advantage of an installation at column vendor site in a lying column (not possible for random packings), further significant cost savings could be achieved by much lower transportation costs, especially when tower and packing vendor are relatively close to each other and column needs to be shipped overseas.

## 4. Conclusions

A heat recovery cooling-condensing design case from a real industrial application was selected to analyse and compare the total investment costs required when considering four different packings. Packings studied include two high capacity plastic random packings, an hybrid plastic structured packing and a conventional metal structured packing. Tower dimensions: diameter and packed bed height to achieve the targeted cooling task were estimated and are significantly different depending on the packing type selected. Lowest investment costs for the four analysed packings is achieved with the plastic hybrid structured packing Hiflow® PLUS#2.

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