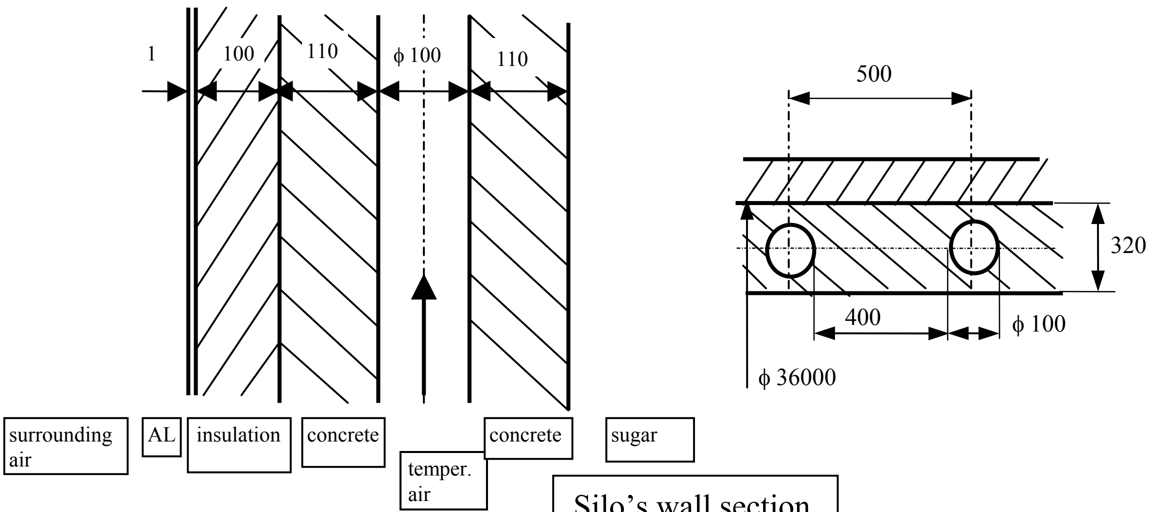
# Determination of heat losses of a concrete silo for sugar and a fan project

VLP - Processing lines of food industry - example, 5/2000

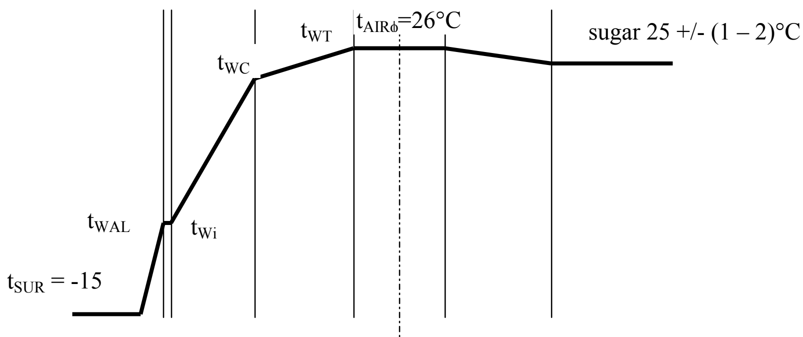
## 1. Given data

| | |
|---|---|
| Silo diameter | D = 36 m |
| Height of a cylindrical part of silo | H = 33 m |
| Concrete wall thickness | T = 0,32 m |

There are 225 holes (vertical “tubes”) with diameter 100 mm and spacing 500 mm for silo’s heat losses compensation in the wall. Tempering air flows upward through tubes (holes), maximal difference between inlet and outlet air temperatures is 4 °C (28 °C → 24 °C). The silo is insulated by a 100 mm thick polystyrene boards and sheathed by a 1 mm thick Al plate.




Silo’s wall section




Temperatures in the silo’s wall

Average sugar temperature in the silo is c. 25 °C. A minimal temperature of surrounding air for calculation is – 15 °C (according Czech standards ČSN

060210 are for Pardubice – 12 °C, Chrudim – 12 °C). In a similar silo were in a tempering air system measured out pressure losses from 2500 to 3500 Pa.

Considering considerably higher heat conductivity of concrete than for sugar and much higher than for the insulation we can suppose that the wall temperature in the wall axis and in a 50 mm distance are the same and are approximately equal to the tempering air temperature. From the same reason we suppose that the wall temperature is practically the same around the silo's perimeter (there is no effect of tubes (holes) spacing).

Another data necessary for calculations:

| | | |
|---|---|---|
| $\lambda_{CON} = 1{,}51$ W/mK | | $s_{CON} = 110$ mm |
| $\lambda_{IN} = 0{,}036$ W/mK | $\rho_{IN} = 30$ kg/m$^3$ | $s_{IN} = 100$ mm |
| $\lambda_{AL} = 209$ W/mK | | $s_{AL} = 1$ mm |
| $\lambda_{SUG} = 0{,}46$ W/mK | $\rho_{SUG} = 770$ kg/m$^3$ | (sugar in bulk) |

Average tempering air temperature is $t_{AIR} = 26$ °C.

According ČSN 730548 is a heat transfer coefficient on an outside vertical silo's wall $\alpha_e = 15$ W/m$^2$K, according ČSN 730542 is in winter (winds) $\alpha_e = 23$ W/m$^2$K. For our calculation we take into account the less advantageous value. The same value we will take for a silo's lid. The values are not important for our calculations, as we will see from following results.

Heat transfer coefficient for heat transfer from the tempering air to the concrete wall of tubes is, according the following calculations, estimated to $\alpha_i \approx 76$ W/m$^2$K. If we do not know a number of tubes we have to estimate an overall heat transfer coefficient k and set the amount of tempering air and a heat flux q. Than we specify an air velocity and consequently $\alpha_i$ and k values. Here is, in brief, state a result of the last iteration (number of tubes and their dimensions are given, amount of the tempering air is specified in the chapter 5.1).

**Specification of the heat transfer coefficient from the tempering air to the tubes wall**

$\lambda_{AIR} = 0{,}065$ W/mK $\quad \nu_{AIR} = 15{,}9*10^{-6}$ m$^2$/s $\quad Pr_{AIR} = 0{,}71 \quad \rho_{AIR} = 1{,}15$ kg/m$^3$

| | | |
|---|---|---|
| Amount of the tempering air | $M_{AIR} = 15{,}6$ kg/s | $V_{AIR} = M_{AIR}/\rho_{AIR} = 15{,}6 / 1{,}15 = 13{,}6$ m$^3$/s |
| Tubes clear area | | $f_T = n * \pi * D_T^2 / 4 = 225*\pi*0{,}1^2/4 = 1{,}77$ m$^2$ |
| Air velocity in tubes | | $w_{AIR} = V_{AIR} / f_T = 13{,}6 / 1{,}77 = 7{,}7$ m/s |

$$Re = w_{AIR} * D_T / \nu_{AIR} = 7{,}7 * 0{,}1 / 15{,}9*10^{-6} = 48428$$

$$Nu = 0{,}023 * Re^{0,8} * Pr^{0,3} = 0{,}023 * 48428^{0,8} * 0{,}71^{0,3} = 116{,}2$$

$$\alpha_i = Nu * \lambda_{AIR} / D_T = 116{,}2 * 0{,}065 / 0{,}1 = 76 \text{ W/m}^2\text{K}$$

## 2. Specification of overall heat transfer coefficient

$$k = \frac{1}{\frac{1}{\alpha_i} + \frac{s_{CON}}{\lambda_{CON}} + \frac{s_{IZ}}{\lambda_{IZ}} + \frac{s_{AL}}{\lambda_{AL}} + \frac{1}{\alpha_e}} = \frac{1}{\frac{1}{76} + \frac{0,11}{1,51} + \frac{0,10}{0,036} + \frac{0,001}{209} + \frac{1}{23}} = 0,32 ... W/m^2K$$

From the equation follows that for a heat loss is determinant the insulation (from 95,6 %). Exact of $\alpha_e$ determination has weigh c. 64 times lower (from c. 1,5 %); as well as is low an effect of the exact tubes location and their spacing on a temperature field in the concrete wall including a determination of a calculated temperature layer (for the of 26 °C) location. The effect of $\alpha_i$ is only from 0,5 % and the concrete wall only from c. 2,3 %. Effect of Al sheathing is c. 580000 times lower.

## 3. Specific heat losses (specific heat flux through composite wall)

$$q_{HL} = k * (t_{AIR} - t_{SUR}) = 0,32 * (26 + 15) = 13,1 \text{ W/m}^2$$

Note: For a wet insulation (mineral wool) were $q_{HL} = 20 - 22$ W/m$^2$.

Note: Silo's wall temperature calculations

| | |
|---|---|
| Inside surface of concrete tubes | $t_{WT} = t_{AIR} - q / \alpha_i = 26,0-13,1/76 = 25,8$ °C |
| Outside surface of the concrete wall | $t_{WC} = t_{WT} - q * s_B / \lambda_B = 25,8-13,1*0,11/1,15 = 24,9$ °C |
| Outside surface of the AL plate | $t_{WAL} = t_{SUR} + q / \alpha_e = -15,0 + 13,1/23 = -14,4$ °C |
| Outside surface of the insulation | $t_{WIN} = t_{WAL} + q * s_{AL} / \lambda_{AL} = -14,4+13,1*0,001/209 = -14,4$ °C |

The calculations confirm our former presumptions about neglecting of the effect of different temperatures in the silo's wall. Simultaneously they show us the effect of separate layers of the silo's wall on the temperatures and consequently on the heat losses too.

## 4. Total heat losses of silo

As a silo's floor is tempered separately we take into account only the surface of the cylindrical part and the lid (by reason of a simplification of the example we will calculate the lid as a circle, not as a conus). We will calculate with an outside silo's diameter (simplifying for worse conditions). Than the heat transfer area and the total silo's heat losses are

$$A_S = \pi * D_S * H + \pi * D_S^2 / 4 = \pi * 36 * 33 + \pi * 36^2 / 4 = 4750 \text{ m}^2$$

$$Q_{HLT} = q_{HL} * A_S = 13,1 * 4750 = 62320 \text{ W} = 62,3 \text{ kW}$$

Note: For an exact calculation of the conical silo's lid (height of the conus H = 9 m) is the area 1137 m$^2$, the circle area is 1018 m$^2$. The relative error is 10,5 %. But if we relate the error to the all silo's surface the relative error is only c.2,5 %. It is negligible. The calculation is shown only as an example what simplifications an engineer can do in practice and what are the errors and their effect to a result (for example an approximate calculation when we do not know an lid shape).

## 5. Calculation of heating air system

### 5.1. Estimation of heating air quantity

Required temperature difference between inlet and outlet air is 4 °C. For the average temperature 26 °C we set

$c_{AIR}$ = 1,0 kJ/kgK $\rho_{AIR}$ = 1,15 kg/$m^3$

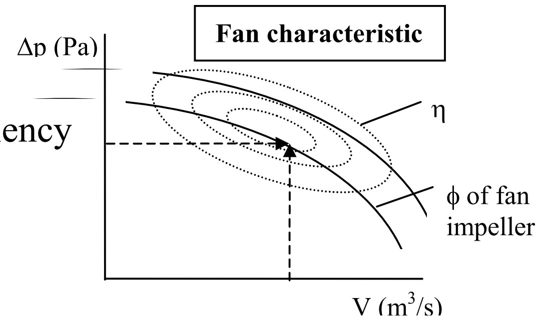
Than is ($Q_{AIR} = Q_{HLT}$)

$$M_{AIR} = Q_{AIR} / (c_{AIR} * \Delta t_{AIR}) = 62,3 / (1,0 * 4) = 15,6 \text{ kg/s}$$

### 5.2. Estimation of fan power requirement

From a calculation of the tempering air system (piping, concrete tubes, channel distribution and collection, butterfly valves atc.) we can set that the total pressure losses in the system are c.

$$\Delta p_L = 3000 \text{ Pa}$$




From a fan characteristic curves we set its efficiency

$$\eta = 0,8$$

Theoretical fan input is

$$P = M_{AIR} * \Delta p_L / (\rho_{AIR} * \eta) = 15,6 * 3000 / (1,15 * 0,8) = 50870 \text{ W}$$

**P = 50,9 kW**

Theoretical electric motor input (power requirement)

$$P_M = 1,2 * P = 1,2 * 50,9 = 61 \text{ kW}$$

### 5.3. Analysis

The value of power requirement is comparable to the heat losses of the all silo. In the option (that was required by a customer) a total energy consumption was c. 62 kW of heat and 51 kW of electric energy. The option is not economical. It is possible to project among others 2 more economical solutions:

- **<u>Installation of an electric heating cables.</u>** Their input is equal to the heat losses. It is not possible to install the system of heating air incl. the fan with its high requirements for energy and an air heater (using steam or hot water).
- **<u>Permit a higher difference between the inlet and outlet air temperatures;</u>** it is the higher difference in the silo's wall temperatures

Both variants are used in praxis. We will test the second variant (as it is more complicated).

## <u>6. Optimised solution</u>

When sugar enters to the silo it has temperature c. 28 °C (the highest sugar layer). During the storage its temperature gradually decrease to c. 23 - 24 °C nearly to a silo's bottom. Here is sugar removed for other processing. Original requirement for the sugar temperature in the silo was 25 +/- (1 - 2) °C, it is from 24 to 26 °C or from 23 to 27 °C. A customer set the stronger limits in the setting. If we take into consideration the wider limit (the temperature of entering sugar is out the limit too) it is possible to admit the higher air temperature difference. In addition it is possible to turn down the air flow. It means from the upward to the downward flow. This change reduces the temperature difference between air and a corresponding sugar layer. The solution makes possible to increase difference between inlet and outlet air temperatures from 4 to 8 °C. In the case the inlet air temperature is 32 °C and the outlet one is 24 °C. Temperatures in the silo are in the next table.

| | Primary solution | | Proposed solution | |
|---|---|---|---|---|
| | Air ↑ | Sugar | Air ↓ | Sugar |
| Top part of silo | 24 °C | 28 °C | 32 °C | 28 °C |
| Middle part of silo | 26 °C | 25 °C | 28 °C | 25 °C |
| Lower part of silo | 28 °C | 23 °C | 24 °C | 23 °C |
| Average difference | 3,3 °C | | 2,7 °C | |

It follows that average differences between air and sugar temperatures in mentioned parts are for both solutions approximately the same. It follows that the solution does not cause problems with moisture migration in the stored sugar (and consequently its hardening). Analogous to the chapter 5., we carry out a new calculation of the tempering air system.

Note: In the middle part of the silo average values are considered.

Regarding temperatures in the silo it is possible to lower air temperature (field experience, good temperature regulation). In the case the inlet air temperature would be c. 30 °C, in the middle part c. 26 °C and outlet air temp. c. 22 °C. For the temperatures would be the heat losses the same like for the primary solution (the same average air temperature i.e. silo's surface temperature). From practice reasons we will calculate with higher temperatures (to see that it is necessary to recalculate heat losses).

## 7. Calculation for optimised option

### 7.1. Specific heat losses

As the average temperature is higher, the heat losses will be higher too. We suppose that the overall heat transfer coefficient will be the same (see note later).

$q_{HL} = k * (t_{AIR} – t_{SUR}) = 0{,}32 * (28 + 15) = 13{,}8\ W/m^2$

### 7.2. Total heat losses of silo

$Q_{HLT} = q_{HL} * A_S = 13{,}8 * 4750 = 65550\ W = 65{,}6\ kW = Q_{AIR}$

### 7.3. Estimation of heating air quantity

In the case is the difference between inlet and outlet air temperatures 8 °C. (32 → 24 °C). For a medium air temperature 28 °C we set

$c_{AIR} = 1{,}0\ kJ/kgK$ $\rho_{AIR} = 1{,}15\ kg/m^3$

Than is

$M_{AIR} = Q_{AIR} / (c_{AIR} * \Delta t_{AIR}) = 65{,}6 / (1{,}0 * 8) = 8{,}2\ kg/s$

### 7.4. Estimation of fan power requirement

The lower circulating heating air quantity the lower pressure loss. Approximately we can say that the loss will reduce in the proportion of squares of air quantities. Than the new pressure loss in the system is

$\Delta p_L = 3000 * (8{,}2 / 15{,}6)^2 = 829\ Pa$

An efficiency of a new fan we set analogous to the chapter 5.2. For example again

$\eta = 0{,}8$

Theoretical fan input is

$P = M_{AIR} * \Delta p_L / (\rho_{AIR} * \eta) = 8{,}2 * 829 / (1{,}15 * 0{,}8) = 7389\ W$

**P = 7,4 kW**

Theoretical electric motor input (power requirement) is

$$P_M = 1{,}2 * P = 1{,}2 * 7{,}4 \approx 9 \text{ kW}$$

## 7.5. Analysis

From the calculations follows that the case makes possible to reduce the fan input from 50,9 kW to 7,4 kW (it is c. 6,8 times). As a price of heat in steam or hot water is much lower than a price of electric energy is the solution more economical than tempering with electric heating cables (even with depreciation of higher fixed assets – fan, heat exchanger, regulating system, piping etc.).

### Approximate economical balance – energy costs

| | |
|---|---|
| Cost of heat in steam for air heating | $C_{HS}$ = 333 Kč/GJ = 1,20 Kč/kWh |
| Cost of electric energy for fan | $C_{EE}$ = 3,00 Kč/kWh |

Primary option – costs per 1 day

$$N_D = (Q_{HLT} * C_{HS} + P * C_{EE}) * 24 = (62{,}3 * 1{,}2 + 50{,}9 * 3{,}00) * 24 = 5459 \text{ Kč/d}$$

Optimised option – costs per day

$$N_D' = (Q_{HLT}' * C_{HS} + P' * C_{EE}) * 24 = (65{,}6 * 1{,}2 + 7{,}4 * 3{,}00) * 24 = 2411 \text{ Kč/d}$$

### Effects of optimised option

$$\mathbf{\Delta N_D} = N_D - N_D' = 5459 - 2411 = \mathbf{3048 \text{ Kč/d}}$$

Note: Because for the optimised option was the amount of tempering air c. 15,6/8,2 = 1,9 times lower than for the primary option, it is in the same ratio lower the velocity in tubes. The result is the lower heat transfer coefficient. For 1,9 times lower velocity is Nusselt number ($Nu \sim Re^{0,8}$) or $\alpha_i$ $1{,}9^{0,8} = 1{,}7$ times lower too. Because the overall heat transfer coefficient (and subsequently heat losses too) is affected by $\alpha_i$ value only from about 0,5 %, its reducing by c. 41 % causes k value reducing only by c. 0.2 %. As the k value was during the calculations round down it is good for nothing to do a new calculations.

Note: Quality of insulation and its sheathing are decisive factors for the total heat losses. When rainwater runs into an insulation its thermal conductivity will be higher and consequently heat losses too (several times). Polystyrene or polyurethane boards are less sensitive than mineral wool.