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PRESSURE LOSSES IN THE NATURAL GAS COOLERS CAUSED BY DEPOSITS ON THE HEAT TRANSFER SURFACES

TLAKOVÉ STRATY U CHLADIČOV ZEMNÉHO PLYNU SPÔSOBENÉ NÁNOSOM NA TEPLOVÝMENNEÝCH PLOCHÁCH

Abstract

Analysed is the effect of the deposits formed on the inside and outside heat transfer surfaces of the natural gas coolers on the pressure losses. Sediments on both sides of the heat transfer surface represent the hydraulic resistance, overcoming of which, for the given degree of gas cooling, requires to increase the compression power and to exploit the higher volume of the cooling air. Operation of the cooler with the clean surface enables to save not negligible amount of the electrical power.

Abstrakt

V článku je analyzovaný vplyv nánosu vytváraného na vonkajšej a vnútornej teplovýmennej ploche u chladičov zemného plynu na tlakové straty. Nános na obidvoch stranách teplovýmennej plochy predstavuje hydraulický odpor, ktorého prekonanie, pri danom stupni ochladenia plynu, vyžaduje navýšiť kompresnú prácu a tiež použiť väčšie objemové množstvo chladiaceho vzduchu. Prevádzkou chladiča s čistými plochami možno ušetriť nezanedbateľné množstvo elektrickej energie.

1 INTRODUCTION

For the natural gas coolers, installed in the compress stations, to achieve the maximum cooling performance, it is necessary for them to be operated free of deposits so on the inside as on the outside heat transfer surface. Maintaining the outer heat transfer surface clean is simpler due to the easy access in comparison to the inside surfaces.

At the present, there is no methodology to assess the degree of the fouling on the inside and outside heat exchange surfaces of coolers in relation to the period of their operation. Fact that such sediments are formed on both heat transfer surfaces is documented in Fig. 1 and 2.

From point of the effect of the deposit on the pressure loss, it can be said that the deposit is the additional resistance increasing the pressure loss. As the problem remains the issue of the deposit formation, intensity and nature of its deposition on the heat transfer surface. It can be hardly assumed, that the deposit would be uniformly distributed so across the section as along the cooler pipes length. More probably this would be a case of the non-uniform fouling on the surfaces and the nature of it may not be exactly defined.

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The overall gas pressure loss represents the total of the local and friction losses. It is necessary to specify the value of the local loss coefficient ζ for each element of the cooler, bound to the concrete location and that is usually rather complicated. When calculating the friction loss, the parameter of the friction factor λ s affected by several factors, mainly by the types of gas flow and pipe surface quality.


a)

b)

Fig. 1 a) Clean inside surface at the inlet section of the gas collector
b) Deposit on the demounted screws if the gas collector

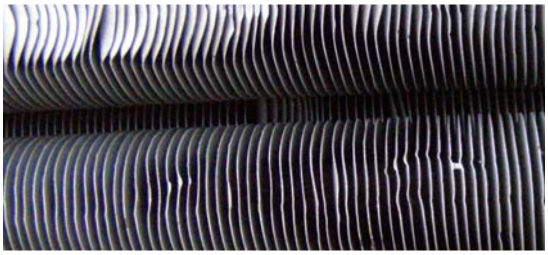
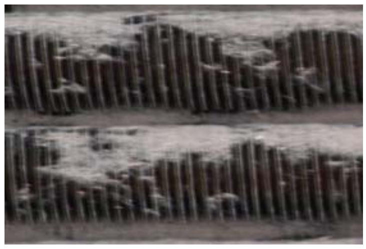

a)

b)

Fig. 2 a) Detail of clean outside heat transfer surface
b) Detail of the outside heat transfer surface by the dust deposit

2 DESCRIPTION OF COOLER

Analysed cooler consists of bundle of pipes with the total number of pipes 1323 pieces, in which pipes are arranged in staggered mode. Indication of the pipes axis spacing as well as the direction of air flow can be seen in Fig. 3. Three rows of pipes in the upper part of the block of the distributor (Fig. 4) direct the compressed gas in one direction and three rows in the lower part of the block in counter direction. Outside surface of the pipe in cooler is finned and the main parameters of the pipe and aluminium fin are given in Tab. 1. The significance of the individual indications in Tab. 1 is obvious from Fig. 5.

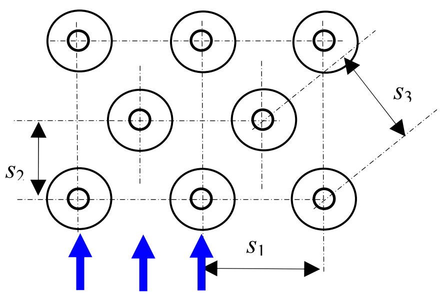


Fig. 3 The arrangements of tubes

Fig. 4 Detail of the cooler block with the pipes arrangement

The cooler assumes the cooling of the compressed gas to the required temperature with the help of seven double-fan units with the air bulk flow in one fan of $Q_V = 41{,}5\ \text{m}^3.\text{s}^{-1}$ and indicated pressure loss with the clean heat transfer surface of 197 Pa.

Tab. 1 Parameters of fan and heat transfer surface

Q_V [m^3.s^{-1}]	Δp [Pa]	b [mm]	s_r [mm]	d_1 [mm]	d_2 [mm]	d_r [mm]	h_r [mm]	$s_1 = s_2 = s_3$ [mm]
41,5	197	2,5	0,65	25	30	58	14	63,5

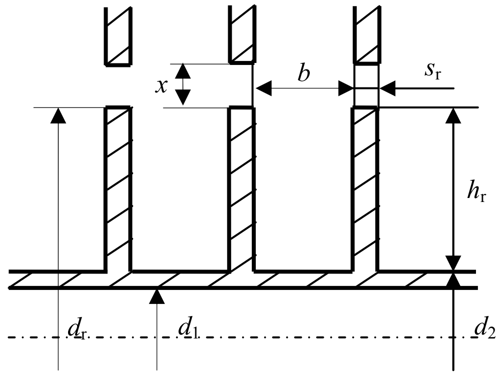


Fig. 5 Geometry of heat transfer surface of the pipe in the cooler

The dirt coming from ambient air during the fan operation is deposited on the outside heat transfer surface. The dirt deposition is more intense on the surface of fins and smooth section of pipes in the lower section of the pipes set of the cooler and significantly wanes in the direction to the last row in the direction of the air flow. Maximum thickness of the deposit on the fins ($h_{n,2}$) is defined by the width of the free space between fins b (Fig. 5). After being filled up, the air flows through the space above the heads of fins with the width marked x and length marked L until the completely filled up.

The total pressure loss in the cooler was determined under the following operational conditions: gas pressure $p = 6{,}53$ MPa, gas temperature at the cooler inlet $t_{1,1} = 60$ °C, gas temperature at the cooler outlet $t_{1,2} = 45$ °C, air temperature $t_{vz} = 20$ °C, gas bulk flow under standard conditions $Q_{V,1} = 1\ 220\ 000\ \text{m}^3.\text{h}^{-1}$. The pressure loss was estimated on the side of air and gas for two alternatives of the heat transfer surfaces quality:

- *clean outside heat transfer surface and the same surface with the deposit thickness marked* $h_{n,2}$,
- *clean inside heat transfer surface and the same surface with the deposit thickness marked* $h_{n,1}$.

3 PRESSURE LOSS ON THE AIR SIDE

According to [1], for the finned surfaces recommended is to execute the calculation of the pressure loss on the air side Δp based on the equation

$$\Delta p = 2 \cdot C_1 \cdot C_2 \cdot (n_r + 1) \cdot Re_{d_h}^{-0,27} \cdot \frac{\upsilon^2}{2} \cdot \rho_2 \tag{1}$$

where
C_1 – correction coefficient depending on the pipes spacing in the direction of the bundle width s_1 [1],
C_2 – correction coefficient depending on the distance of the pipes centre s_3 [1],
υ – rate of air flow in the narrowest cross section [m.s^{-1}],
ρ_2 – air density at the mean air temperature [kg.m^{-3}],
n_r – number of pipes rows in the air flow direction [1],
Re – Reynolds number calculated for the hydraulic diameter d_h [1].

Reynolds number is determined from the equation

$$Re_{d_h} = \frac{\upsilon \cdot d_h}{\nu} \tag{2}$$

where
υ – air flow rate in the narrowest cross section [m.s^{-1}],

ν – kinematic viscosity of air [m^2.s^{-1}].

The hydraulic diameter for the clean cooler surface can be calculated from the equation (3)

$$d_h = \frac{4 \cdot S_p}{O} = \frac{2 \cdot [(s_1 - d_2) \cdot b + s_r \cdot x]}{(2.h_r + b + s_r)} \tag{3}$$

where the significance of the individual indications follows out from the Tab. 1. Correction coefficients C_1 and C_2 are derived based on the arrangement of the pipes in the cooler and their geometry from the graphs given in Fig. 6 and 7.

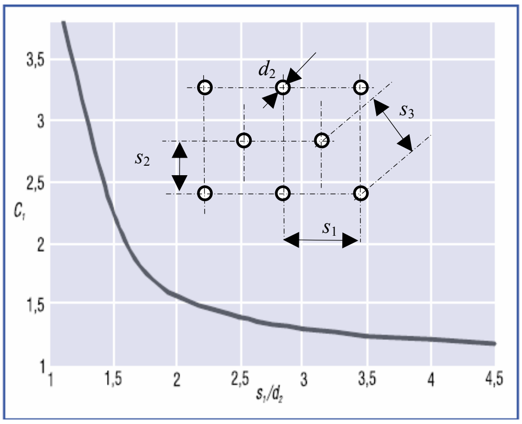


Fig. 6 Correction coefficient C_1

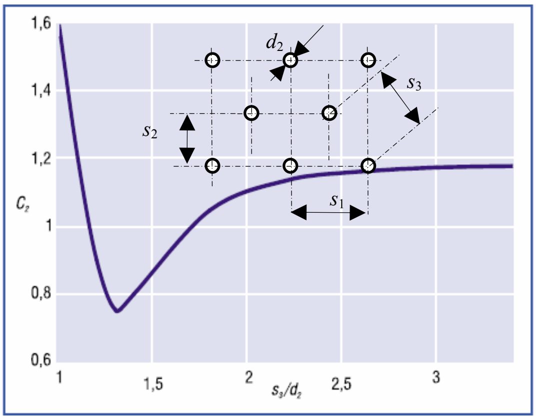


Fig. 7 Correction coefficient C_2

The pressure loss (in Pa) on the air side can be obtained according to [2] from the simplified equation

$$\Delta p = \zeta \cdot n_r \cdot \frac{\upsilon^2}{2} \cdot \rho_2 \tag{4}$$

where

ζ – local pressure loss coefficient depending on the pipes bundle geometry and Reynolds number determined for the equivalent diameter d_e [1] (more details in [2]).

n_r – number of rows of pipes in the air flow direction [1],

υ – flow rate in the narrowest cross section in the bundle [m.s^{-1}].

All information's from the accessible literature stress the fact that the existing equations, recommended for the calculation of the pressure loss on the finned pipes, provide rather different results and that the calculation of the pressure loss value realised based on those, is the informative only. The confrontation of the values of the pressure loss obtained from calculation may be executed comparing them with the results obtained from experiments or applying the numerical simulation for the given geometry of the heat transfer surface [4].

For the cooler CH_4 in KS01 in Veľké Kapušany, the obtained course of the pressure loss according to Dvořák [1] and Cikhatr [2] as depending on the cleanness of the outside heat transfer surfaces, is presented in Fig. 8. For he clean outside surface of fins, is the pressure loss calculated according to [1] equal to 46 Pa and according to [2] equal to 59 Pa. The difference between the values of the pressure loss according to both literatures represents 28 %. The increasing thickness of the deposit on the fins causes the rise in the pressure loss and for the deposit thickness about $h_{n,2} = 0{,}84$ mm, on each side of the fin, it gains the value denoted on the shield of the fan - 197 Pa. The complete filling up of the space between the fins occurs for the deposit with the thickness $h_{n,2} = 1{,}25$ mm. In such case of the outside surface fouling, the pressure loss would achieve about 750 Pa, and this exceeds much more the capacity of the installed fans.

The pressure loss on the side of gas depends first of all on the quality of the inside surface covered by deposit and on the gas flow rate. The minor pressure loss has to be estimated for each type of the resistance individually. The equations taking into the account the changed properties of the gas due to its actual pressure and temperature are applied when calculating the pressure loss frequently. When calculating the major as well as minor pressure losses, applied are further modifications of the basic equations. For the high pressure and medium pressure pipelines, for the pressure loss calculation applied may be such equations, which are derived from the general expression of the second power of the pressure difference on the length L. After the principal equations modification, the pressure loss in the cooler may be expressed as (5)

$$\Delta p = p_1 - \sqrt{p_1^2 - \left(\sum_{i=1}^{n} \lambda_i \cdot \frac{L_i}{d_i} \cdot \frac{z_s \cdot r \cdot T_s \cdot Q_m}{S_i^2} + \sum_{j=1}^{m} \xi_j \cdot \frac{z_s \cdot r \cdot T_s \cdot Q_m}{S_j^2} \right)} \qquad (5)$$

where

p_1 – gas pressure in front of the cooler [Pa],
p_2 – gas pressure at the outlet from cooler [Pa],
z_s – mean value of the compressibility factor [1],
Q_m – muss flow of gas [kg.s^{-1}],
T_s – mean gas temperature in the cooler [K],
r – gas constant [J.kg^{-1}.K^{-1}],
S_i, S_j – cross section area in the concrete location of the cooler [m^2].

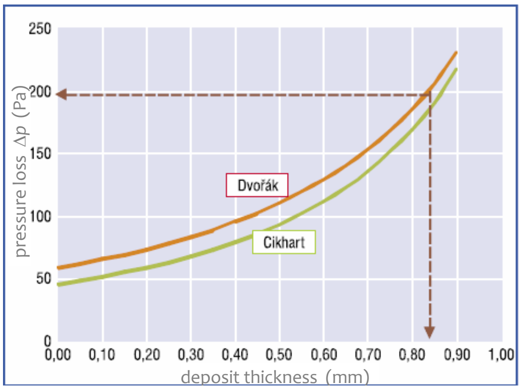


Fig. 8 Pressure loss as depending on the cleanness of the outside heat transfer surface

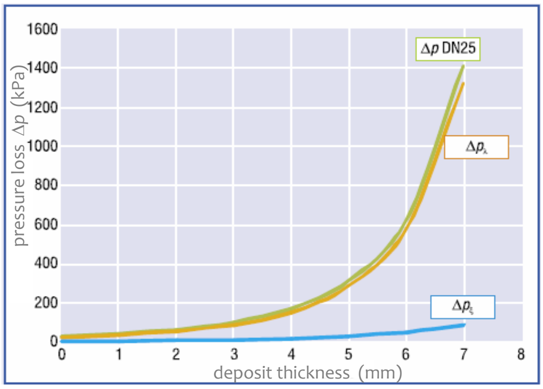


Fig. 9 Pressure loss as depending on the cleanness of the inside heat transfer surface

There are the pipes with three different nominal diameters (DN25, DN125, DN300) in the analysed cooler. The cross section of the pipe with the inside diameter of 25 mm will be reduced by 8 % due to the deposit 1 mm thick. This reduction in the cross section will cause the rise in the gas flow rate by 18,1 %, and this will then cause the rise of the friction pressure loss by about 39,6 %. In case of other pipes installed in front of and behind the system of the cooling pipes, the effect of the same thickness of the deposit will contribute to the flow rate rise less significantly. For the pipes with the diameter of 125 mm, the flow rate will be increased by 3,3 % (pressure loss by 6,7 %). For the pipes with the diameter of 300 mm, the rise in gas flow rate will be increased by 1,3 % and pressure loss by 2,7 %. The effect of the deposit on the inside surface will be more significantly manifested for its greater thickness (Fig. 9). For the thickness $h_{n,1} = 7$ mm, the total pressure loss will be 1452 kPa, which means even 22,2 % out of the value of the operational pressure. The most significant is the pressure loss due to the friction on the pipe with the nominal diameter DN25. This participates in the overall friction pressure loss in case of the analysed cooler even with 99,5 %. The value of the friction pressure loos on the pipe DN25 represents even 91 % out of the total pressure loos.

4 CONLCUSION

If considered is the economy of the natural gas supply in the individual compressor stations; from point of the pressure loos justified is the cleaning of the inside heat exchange surfaces and this should be executed in the regular intervals. Then deposits with the thickness greater than 3 mm would not be formed on the inside surfaces of the pipes and the total pressure loos would be about 2 % out of the operational pressure which is an acceptable level. The need to increase the compression power with such deposit thickness in comparison to the cooling systems with clean surfaces, is about 7,5 %.

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