Cooling of Compressed Gas with Thermosyphon Heat Exchange

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Cooling of Compressed Gas with Thermosyphon Heat Exchange: *A system is developed that is comprised of finned tube termosyphons, which are used for cooling the natural gas pipelines. On the basis of heat balance equations a mathematical model is developed to calculate the geometric dimensions of the thermosyphon's elements. A feasibility assessment of the appropriateness of implementing the system is also conducted.*

Key words: *Cooling systems, Cooling of compressed natural gas, Termosyphons.*

INTRODUCTION

The Bulgarian gas transmission network is built in a ring-shaped form consisting of high-pressure gas pipelines with a total length of 1700 km, three compressor stations with total installed capacity of 49 MW. The transit gas transmission network comprises high-pressure gas pipelines of total length of 945 km with prevailing diameter of DN 1000, six compressor stations with total installed capacity of 214 MW [4].

Cooling of natural gas in the main pipelines after the compression process in the responsive compressor stations is done by means of air-cooling machines. The installed capacity of the electrical generators in one of those machines ranges from 60 to 110 kW, and the total power of one compression station varies between 1200 and 2200 kW [1]. A number of constructive solutions, dealing with air-cooling machines, are available. They are all based on the transported gas' power, its temperature, as well as the environment temperature [1,2].

In practice, after cooling in the aforementioned machines has taken place, the natural gas' temperature is higher by approximately 15-20^{0}C than the external air, which is supplied by fans in order to be cooled [1,2]. A deviation of the projected values in the cooled transported gas' temperature of only 3-4^{0}C conduces to a gas overrun of about 1% [1]. When the gas temperature at the end of the air-cooling systems surpasses 30^{o}C, the fans' engines are automatically turned on. Stemming from exploitation analysis of air-cooling machines, an increase in reliability and effectiveness of those machines may be reached by means of gradually putting in motion the ventilators' electrical motors, and also by refining the diagnostics and technical utilisation of the electromechanical equipment. Nevertheless, none of the hitherto discussed methods provides a solution to the issue of energy overrun, which is interlinked to the exploitation of the electrical motors of the air-cooling machines [1].

CONSTRUCTION OF SYSTEM FOR COOLING OF COMPRESSED NATURAL GAS

A thermosiphon system for cooling of main pipelines is suggested beneath (fig.1), with the help of which the pipeline's heat is released into the atmosphere. During this process the heat exchange taking place in the air-cooling machine intensifies. In this particular system the heat exchangers are placed in a frame, filled with cooling agent, which boils at a temperature lower than the cooled gas' one at the entry point of the cooling machine.

The velocity, which the cool air moves at in the space between the pipes of the cooling system's coating, is a function of the geometrical measurements of the thermosiphon (the breadth and length of the pipes, the size of the finned part), both the environment's, and the compressed gas' temperatures, the pressure drop from the ventilator and the temperature difference of the air at the entry and exit points of the apparatus.

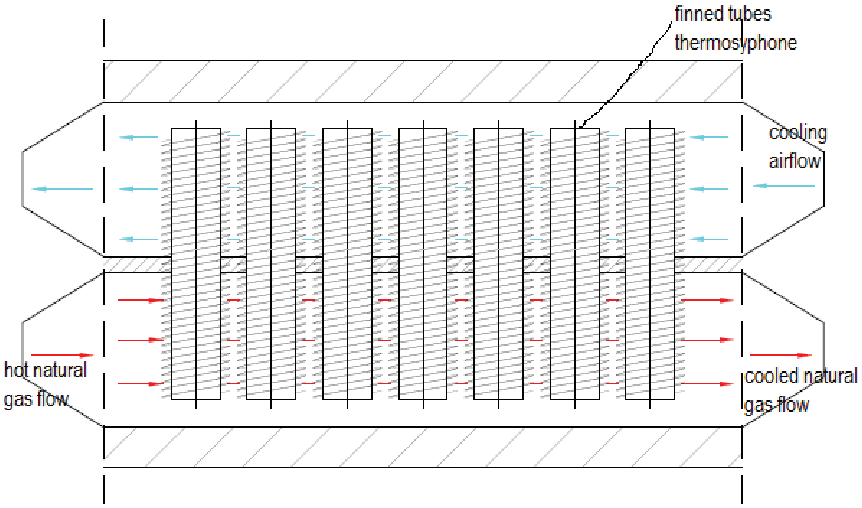

Fig.1. General view of the main pipeline cooling system

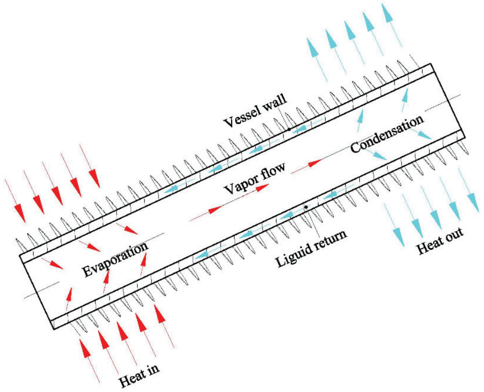

Fig.2. Construction of heat pipe

METHODOLOGY

The thermal calculations of the pipeline cooling system are based on the subsequent equations herein:

- Equation of the thermal balance during the heat transfer ensuing from the natural gas to the thermosiphons' heated surface:

$$\dot{m}_g c_{pg}(t'_g - t''_g) = \alpha_g F_g (t_g - t_w) + m_{wHE} c_{pwHE} \frac{d}{d\tau} t_{wHE} \qquad (1)$$

where: $\dot{m}_g$ - mass flow of cooled gas, kg/s; c_{pg} - specific heat capacity of the gas at p=const, J/(kg.K); t'_g, t''_g - gas temperature at the entry and exit points of the heat exchanger respectively, ^{0}C; α_g - heat exchange coefficient from the gas in the direction of the heat exchanger's pipes, W/(m^{2}K); F_g - heated surface of the pipes from the gas side, m^2; m_{wHE} - the mass of the heat exchanger's pipes, kg; c_{pwHE} - specific heat capacity of the heat exchanger's pipes, J/(kg.K); t_{wHE} - the average temperature of the pipes, which the heat exchanger is comprised of, ^{0}C;

- Equation of the thermal balance, occurring from the internal surface of the thermosiphon to the evaporating fluid:

$$\alpha_g F_g (t_g - t_{wHE}) = \alpha_{evap} F_f (t_{wHE} - t_f) + m_{HE} c_{pHE} \frac{d}{d\tau} t_{wHE} \tag{2}$$

where: α_{evap} - heat transfer coefficient from the pipes' sides to the liquid fluid, W/(m^2.K); F_f - heat exchange surface of the pipes to liquid fluid, m^2; t_f - the temperature of the liquid fluid, ^{0}C;

- The thermal energy that the condensing steam carries on to the internal surface of the thermosiphon is utilised for increasing the temperature of the latter's construction, as well as for air heating.

$$\alpha_{cond} F_{cond} (t_{film} - t_w) = (m_w c_w + m_P c_P) \frac{d}{d\tau} t_w + \alpha_{air} F_{air} (t_w - t''_{air}) \tag{3}$$

- Equation of the thermal balance at the air being heated from the thermosiphon:

$$\alpha_{air} h.(\pi d + 4a)(t_{wHE} - t_{\mathrm{air}}) = c_{pair} \dot{m}_{\mathrm{air}} (t'_{air} - t''_{air}) + m_w c_{pwHE} \frac{d}{d\tau}(t_{wHE}) \tag{4}$$

where: α_{air} - heat transfer coefficient from the thermosiphon's surface to the air, W/(m^2.K); $m_{\hat{A}}$ - air mass in the coating, kg; c_{pair} - the specific heat capacity of the air within the coating, J/(kg.K), $\dot{m}_{\mathrm{air}}$ - air's mass flow, kg/s; t'_{air} - air temperature at the inlet of the coating, ^{0}C; t''_{air} - air temperature at the outlet of the coating, ^{0}C; *h*- height of the ventilation shaft, m; *d* – external diameter of the thermosiphon, m; *a*- fin length, m;

The thermal calculations herein use the following initial data: *D*, m – outer pipe diameter; *a* - m, length of the interlinking pipe's separator; t'_{air}, °C – inlet air temperature (established as 0°C); t_{wHE},°C – temperature of the pipe's side (established as being equal to the gas' temperature); t''_{air},°C – outlet air temperature at the apparatus; *R*, J/(mol.K) – universal gas constant; *g*, m/s^2 – acceleration during free fall; P_a, *Pa* – atmospheric pressure; *M* - kg/mol, molar mass of air.

Sequence of thermal calculations

At a set ventilation shaft height, *h*, m the following calculations are carried out:

- We establish the air density both at the inlet and outlet of the ventilation shaft:

$$\rho_{air} = P_a \cdot \frac{M}{R \cdot T_{air}}, kg/m^3 \tag{5}$$

- The equivalent diameter of the ventilation shaft can be express with:

$$d_{eqv} = \sqrt{\frac{\left[(a+d)^2 - 3.142 \cdot \dfrac{d^2}{4}\right] \cdot 4}{\pi}}, m \tag{6}$$

- Air velocity in the shaft is determined using the method of subsequent approximations, whereas the Reynold's number is established as convergent criteria.
- Air pressure drop in the heat exchanger is evaluated using the following formula:

  ○ At natural convection: $\Delta P = h \cdot g \cdot \dfrac{(\rho' - \rho'')}{2}, Pa$ (7)

○At forced convection: $\Delta P = h \cdot g \cdot \frac{(\rho' - \rho'')}{2} + \Delta P_{air\,fan},\ Pa$ (8)

- Air velocity can be determined by the following equation:

$$V_{air} = \sqrt{2 \cdot \frac{\Delta P}{\chi \cdot \frac{h \cdot \rho_{av}}{d_{\mathrm{eqv}}}}}, m/s \tag{9}$$

where the major loss coefficient can be estimated by Blausius' formula

$$\chi = \frac{0.3164}{\mathrm{Re}^{0.25}} \tag{10}$$

- The specified Reynolds number

$$\mathrm{Re} = \frac{V_{air} \cdot d_{eqv}}{\nu_{air}} \tag{11}$$

- The air flow regime is determined by the value of Reynolds number (at laminar, transitional and turbulent flow);
- After a criterial equation has been selected, the heat transfer coefficient α has to be calculated as well, by means of the Nusselt number; formulas for fin tube thermosyphon are used.
- From the equivalency condition between the input and the output energy of the ventilation shaft we derive at an equation useful for assessing the height of the aforementioned shaft:

$$h = \frac{N_{out}}{\left[\alpha \cdot (\pi \cdot d + 4 \cdot a) \cdot \left(T_w - \overline{T}_{air}\right)\right]} \tag{12}$$

On the basis of the presented methodology, calculations for a compressor station have been carried out, which allows for an assessment of the investments, electrical energy savings, the payback period, NPV and IRR of the installation for cooling the main natural gas pipelines.

The standard cooling systems can lessen the natural gas' temperature from 600 ^{o}C to 300 ^{o}C (at natural gas pressure after compression being 5.4 MPa), however during the summer period the cooling temperature cannot be reached, thus that requires the ventilators to be constantly loaded. In practice, they are not shut down during the duration of 4-5 months.

A comparative analysis has been carried out at an already existing cooling system of a main natural gas pipe in the town of Petrich. The system comprises of 4 fans with a power of 40 kW and puts to use a new method using a finned tubes thermosiphons heat exchanger for cooling. The volume flow of the cooled natural gas is 342466 m^3/h. The power required for suction and discharge of warm air is 3.2 times less than the power used during the standard cooling process, in part due to the greater efficiency of the fin-tubes thermosiphones' efficiency at cooling natural gas.

CONCLUSIONS

1. On the basis of the represented methodologies a mathematical model used for calculating the optimal geometry of a heat exchanger with finned tubes thermosiphons has been developed.
2. The calculations conducted exhibit that the temperature reduction of natural gas through one section is approximately 24-32^{0}C. At 30^{0}C the ventilation system automatically switches itself off.
3. The proposed installation for cooling of main pipeline has a low payback period of 1.5 years.

4. The propounded method leads to energy savings due to the reduced installed power capacity of the ventilator.

5. When using suitable heat and electricity consumers, as well as at higher pressures at the compressor station (the gas storage facility in the village of Chiren, Bulgaria), a cogeneration installation utilising an organic Rankin cycle would be beneficial [2].

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This paper has been reviewed.