Thermal Energy Loss in the Steam Valves and its Effects

Galip Temir and Durriye Bilge
Mechanical Engineering Faculty,
Yildiz Technical University, 34349 Besiktas, Istanbul, Turkey

Abstract: In this study, the effects of thermal losses in steam valves for flow have been analyzed. In a set of experiment built to achieve this, pressure and condensate values have been calculated in different temperatures for insulated and non-insulated cases in three different nominal diameter steam valves. The amount of energy lost from the valve surface in the natural convection conditions has been calculated through application and use of principles of heat transfer and fluid mechanics and equations of mass and energy conservation. Thermal loss coefficient change and the amount of the heat transferred from surface in 150°C steam temperature for different steam velocities have been presented in diagrams. Heat losses have been calculated in two different steam velocities in the cases of five different steam temperatures and their diagrams have been arranged.

Key words: Steam Valves, Thermal Energy, Steam Temperature

INTRODUCTION

Pressured pipelines used in various industrial areas such as textile, leather, chemistry, medicine are insulated in order to prevent thermal losses. However, insulation could be neglected in some components such as valves and filters due to the ignorance for the thermal losses at these parts. At these parts, steam goes into throttling process and a temperature drop occurs in the water steam which is the real gas. In addition, loss of enthalpy occurs related to the thermal loss. Consequently, the steam arrives saturation line and a two-phase flow emerges in the system. This case causes a deterioration of flow conditions in the system, an increase in pressure drops and a rise in the convection heat transfer coefficient because of the condensation process. Steam energy could not be used to its optimal capacity. Steam Traps often have to become active in order to stabilize the flow balance in the system [1]. A more extensive power is needed in order to realize the desired operating conditions. The experiment set has been built in a factory producing steam equipments in Izmit, Turkey. Valve pressure losses and thermal energy losses in the insulated and non-insulated valves have been calculated. In the experiments, DN25, DN50, DN80 cast iron globe valves were used in different steam temperatures, mass flow rates and velocities ranging to 20 m/s.

Theoretical study: As the steam passes through the valve, it goes into throttling process. The effect of change in temperature of an isenthalpic change in pressure is represented by the Joule- Thomson coefficient m, defined by:

$$\mu = \left(\frac{\partial T}{\partial P} \right)_h \tag{1}$$

where the derivative is interpreted as the change in temperature due to a change in pressure at constant enthalpy[2]. In the throttling process temperature is stable as far as the ideal gases are concerned. Joule-Thomson coefficient of water steam is positive and when it is throttled, its temperature drops. If the valve is not insulated well against the heat, the process does not become adiabatic due to the fact that the heat will be transferred out and temperature and enthalpy also drop down. Work loss in adiabatic throttling is[3]:

$$W = m_b . T_o . (s_1 - s_2) \tag{2}$$

The amount of the heat transferred out from the surface of the valve in the throttling process is (4):

$$Q = \alpha . A . (T_s - T_o) \tag{3}$$

The total decrease in the energy of the steam is:

$$E = W + Q \tag{4}$$

In the throttling process, the work loss defined in the equation 2 cannot be changed, thermal loss Q_L can be reduced through insulation. When the valve is fully turned on, pressure drop can be calculated as follows [5]:

$$\frac{\Delta P}{d} = k \frac{Q^2}{2g} \tag{5}$$

Enthalpy change for real gases is[6]:

1. Boiler   2. Collector   3. Heat exchanger   4.Tested valve
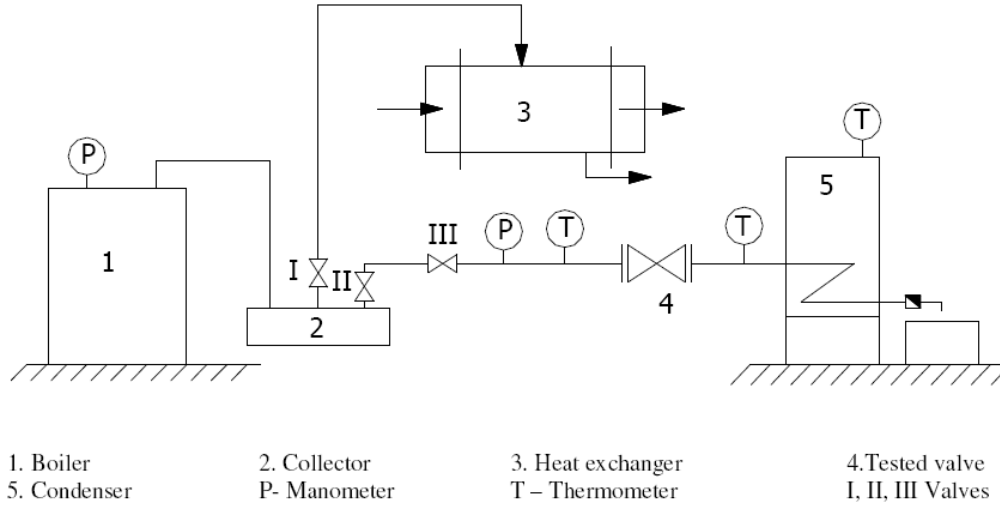5. Condenser   P- Manometer   T – Thermometer   I, II, III Valves

Fig. 1: Experimental Set-up of the Studied System

$$dh = c_p.dT + \left[v - T\left(\frac{\partial v}{\partial T}\right)_p \right] dP \qquad (6)$$

There is not any enthalpy change in the throttling process:

$$\Delta h = \int_{T_1}^{T_2} c_p dT + \int_{P_1}^{P_2} \left[v - T\left(\frac{\partial v}{\partial T}\right)_p \right] dP = 0 \qquad (7)$$

In this equation, in order to solve $\int_1^2 \left[v - T\left(\frac{\partial v}{\partial T}\right)_p \right] dP$ integral, $\left(\frac{\partial v}{\partial T}\right)$ change should be known.

The enthalpy change in the real gases could be found out through the equation below using generalized enthalpy departure chart[3]. Here, (*) denotes an ideal gas state:

$$h_2 - h_1 = (h_1^* - h_1) + (h_2^* - h_1^*) - (h_2^* - h_2) \qquad (8)$$

In the equation 8, reduced pressure and reduced temperature values are used for the inlet and outlet states of the steam in the valve. As the outlet temperature is not known, the result is gained through trial and error method. The entropy change, which occurs as a result of adiabatic throttling, is calculated by the equation below sourced from generalized entropy departure chart diagram according to the outlet temperature value found out with the equation 8:

$$s_2 - s_1 = (s_2^* - s_2) + (s_1^* - s_2^*) + (s_1^* - s_1) \qquad (9)$$

If steam inlet and outlet states are clear in a fully turned on valve, the total amount of energy which steam losses are:

$$E = m(h_i - h_e) \qquad (10)$$

Here, $(h_i - h_e)$, can be found as follows through equation 8:

$$h_i - h_e = (h_i^* - h_i) + (h_e^* - h_i^*) - (h_e^* - h_e) \qquad (11)$$

Heat transferred out from the valve is calculated through equation 3 if the surface temperature of the valve is known. If it is not known, the transferred heat can be calculated through the equations 10 and 4.

Experimental set-up and study: Schematic layout of the experimental setup has been given in Fig. 1. The steam taken from the boiler comes to the collector, passes through the valve to be tested and is condensed in the outlet of the valve. Some of the steam is taken into the heat exchanger and condensed and let go to the condense tank in order to adjust the steam mass flow rate.

In the experiments, studies have been realized in saturated steam with pressure ranging between 5 bar – 8 bar or in superheated steam with temperature ranging between 120–160°C because the steam needs of small and medium scale industrial plants are expected to be in these ranges.

Saturated steam coming out of the boiler in 5 bar pressure has been exposed to throttling in 0.5, 1, 2, 3 and 4 bar pressure respectively in the outlet of the collector and the superheated steam has been let to come into the valve to be tested. Steam velocity has been arranged as 3 m/s and 20m/s through the heat exchanger. Pressure values in the inlet and outlet of the valve have been measured out with a Pakkers manometer with 0.5 bar scale division and (0-10) bar pressure range. A NiCr-Ni, metal cable, K type,

Tecmoterm thermocouple with (-120°C, +1300°C) scale division, F0.2 sensibility and 0.1°C scale division has been used in order to measure the temperature. Condensed steam mass flow rate has been measured with a digital Boster scale with 2.0 gr caliber and 0.30kg capacity.

Arranging the diagrams and results: The pressure difference in the valve inlet and outlet could not be observed with existing manometers. The out coming pressure difference has been found out using loss coefficient taken from the diagram providing diameter versus friction loss coefficient which has been arranged for Globe valves Fig. 2[5] through the equation below:

$$k_p \frac{c^2}{2g} = \frac{\Delta P}{d} \tag{12}$$

Pressure losses found out are in the range of 28.5 Pa and 200 Pa. These values have been used in the equation 5 and lost energy has been calculated. The outlet temperature of the steam in the valve has been found with the equation 8 through the trial and error method. Lost work has been found with the equations 9, 10, 4 and 2. The total amount of lost energy and heat loss transferred out have been found. During the experiments, it has been observed that change in pressure and temperature (ranging from 0.2 to 4.5°C) is considerably small and specific volume of the steam in the range of study has been assumed constant. Using the equation of mass conservation, velocity of steam c has been found:

$$c = \frac{4m}{\pi D^2} \tag{13}$$

The enthalpy of the steam in the inlet of the valve has been calculated:

$$H = mh \tag{14}$$

The amount of the heat loss found out with the equation 4 has been divided by the entropy value found out with the equation 14 and dimensions thermal loss coefficient has been calculated:

$$k = \frac{Q}{H} \tag{15}$$

The velocity of the steam in different steam mass flow rates and temperatures, the amount of the heat transferred from the steam to the surroundings and the values of heat loss coefficient have been calculated. Hence, for the valves insulated and non-insulated:
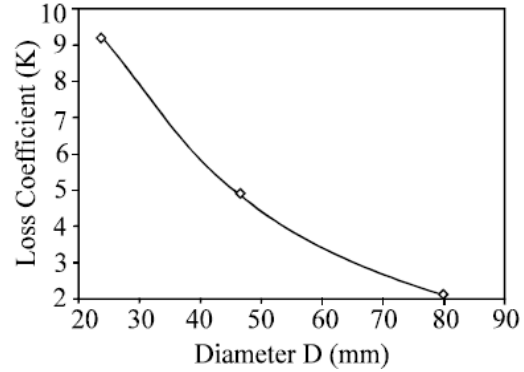

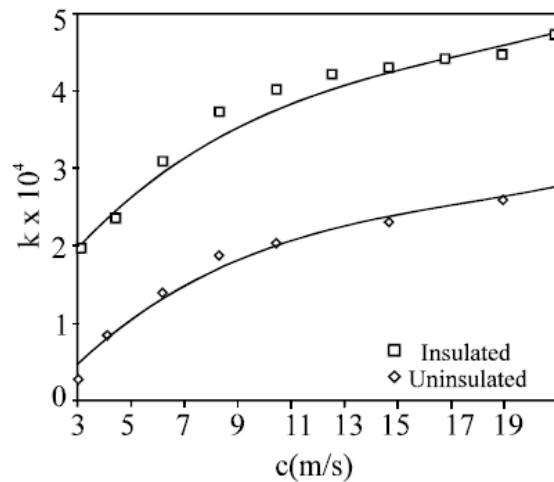Fig. 2: Diameter versus loss coefficient for globe valves (4)

Fig. 3: Thermal loss coefficient versus steam temperature variation in 150°C temperature for DN 25, DN 50, DN 80 Valves
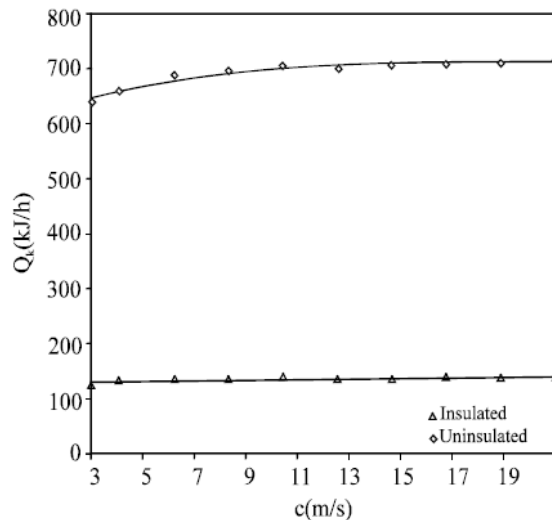

Fig. 4: Lost heat versus steam velocity at 150°C temperature in DN 80 valve
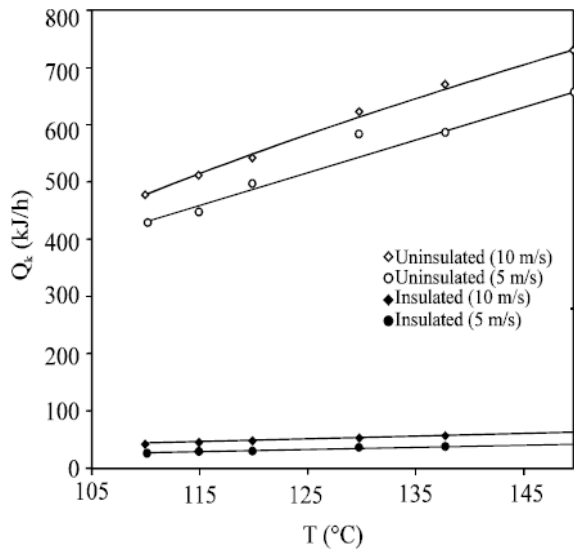
Fig. 5: Heat loss versus steam temperature variation of constant steam velocity in DN 80 cast iron globe valve

a) The diagrams of thermal loss coefficient versus the steam velocity (Fig. 3) and the lost heat versus the steam velocity in the constant steam temperature have been arranged (Fig. 4).

b) The diagrams of heat loss versus the steam temperature in the constant steam velocity have been arranged (Fig. 5).

RESULTS AND DISCUSSION

1. It has been found out that as the velocity increases, the increase of the thermal loss coefficient becomes lower after a definite change in velocity and that, in the insulated case, this value could be obtained in lower velocities (Fig. 3 and 4).

2. The change in heat loss related to the steam temperature in the constant steam velocities has exhibited nearly a linear character (Fig. 5).
   Compared with non-insulated cases, increases in temperature and heat loss are lower with insulated cases.

3. As seen in Fig. 4, the difference between the inlet and outlet temperatures of the steam is reduced with the use of insulation.

4. The experiment has been applied in the 25.6°C dry bulb room temperature sending air in 2.5 m/s velocity vertical to the horizontal valve axis. The average surface temperature of the valve has been found out as 122°C in the non-insulated valve, 140°C steam temperature and 10 m/s steam velocity. The heat transfer coefficient has been found through the connection between valve external surface and room as 18 W/m^2K using the known methods under these conditions. When the values found out are used in the equation 3, heat loss has been calculated as 593.43 kJ/h. This value can be found out with diagram 2 or equation 4 as 642 kJ/h.

5. The decrease in the steam temperature and pressure at the end of the throttling causes the steam density to become lower and the steam velocity increase. As the dimensionless Reynolds number is analyzed, the change in the Re value is proportionate with the multiplication of velocity and density. Therefore, the increase in the velocity of heat transfer drops after reaching to a particular value through the connection between the steam and the internal surface of the valve. This decrease is confirmed with the drop of the velocity of the increase of the loss coefficient k and heat loss after reaching to a particular value in the diagrams of Fig. 3 and 4.

As the fact that almost 2.200.000 valves operated without insulation in Turkey in the last ten years is taken into consideration, a heat loss of 642kJ/h per hour in the steam temperature of 140°C emerges. This value can be calculated for one valve as follows:

642 * 20 * 300= 3,852,000 kJ/year

with a hypothesis that an industrial plant operates 20 hours a day, 300 days in one year.

Therefore, as far as the approximate amount of 2,200,000 valves is concerned, the loss can be calculated as follows:

2,200,000 x 3,852,000= $8.47*10^{12}$kJ/year

The cost of 1 kg fuel oil spent for operating a boiler with average 80% efficiency is 81,265,000$/year. In this sample handled, the pressure has been assumed as 4 bars while the temperature has been assumed as 140°C. Since the steam pressure and temperature increases are parallel with energy losses insulation maintains its vitality.

The steam may reach its saturation line and this may cause a two-phase flow because of the heat losses from the valves[7]. The clarification of the location of steam traps, which must be placed on piping elements in order to prevent the deterioration of flow character, is also worth attention.

Nomenclature

A   surface area, m^2
C   velocity, m/s
cp   constant pressure specific heat, kJ/kg K
D   internal diameter, m
d   density, kg/m^3
g   gravitational acceleration, m/s^2
h   specific enthalpy, kJ/kg
k   dimensionless thermal loss coefficient
kp   friction loss coefficient
m   mass flow rate, kg/s
P   pressure, kPa, bar
Q   heat, kW
s   specific entropy, kJ/kg/K
T   temperature, °C, K
W   power, kW
v   specific volume, m^3/kg
μ   Joule Thomson coefficient, K/kPa

Subscripts
e   outlet
i   inlet
o   ambient
s   surface

REFERENCES

1. Thermo-Steam Equipments Catalogue, 1996. Thermo Steam Equipments End. Company, (in Turkish).
2. Maron, H. and T.B, Lando, 1974. Fundamentals of Physical Chemistry. Macmillan Publishing Co. Inc. New York.
3. Cengel, Y. And M. Boles, 1989.Thermodynamics, an Engineering Approach. Mc-Graw-Hill, New York.
4. Incorporeal,. F. P. And. D. P., DeWitt, 1985. Fundamentals of Heat and Mass Transfer. John Wiley and Sons. New York.
5. Gentez, I. And M. Sogukoglu, 1993. Calculation of Local Energy Loss Coefficient of Valves. Thermodynamics, 5: 42-48.
6. Kirillin, V.A., V.V. Sychev, A.E. Sheindlin, 1976. Engineering Thermodynamics, Mir Publishers, Moscow.
7. Nagel, H., 1986. Analysis of Flow Induced Value Operation and Pressure Wave Propagation for Single and Two-phase Flow. Nuclear Engineering and Design, 96: 115-121.