WHEN YOU NEED TO BE SURE

CyFlex® Knowledge Article

Natural Gas Burner Sizing Calculations

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Background

The Servotech emissions abatement system uses a natural gas burner to help heat the exhaust stream so the SCR can operate in its most efficient temperature range. Under certain engine operating conditions, it appears that the burner is undersized for this task. Given the necessary inputs, it should be possible to perform real-time calculations of the natural gas flow rate that would be required to achieve the target SCR bed temperature. This paper is intended to document those calculations.

Natural Gas Properties

The facility natural gas composition is continuously monitored by an Online Natural Gas Analyzer (ONGA). A central CyFlex node monitors the ONGA device for the most recent update and transfers the composition data to a central database and to all CyFlex test cell nodes that have requested it. The `update_composition` task at the test cell uses that information to update the natural gas composition vector, `ngC.` to reflect the most recent values. The `gas_prop` task computes a variety of natural gas properties for the measured composition at a given temperature and pressure.

The parameters of primary interest for these calculations are the molar mass, `ngC.MM`, the lower heating value by mass, `ngP.LM`, and the wet stoichiometric air/fuel ratio, `ngP.RW.`

Adiabatic Combustion

The lower heating value is the amount of energy that will be released when a unit mass of natural gas is burned with air and the resulting combustion products are returned to the initial temperature of the reactants. For the burner in question, the burned gas mixture is not returned to a low temperature but is instead mixed with the engine exhaust to raise its temperature. We can calculate the maximum adiabatic temperature of the burner exhaust if we know the combustor air/fuel ratio. The maximum temperature will occur when the air/fuel ratio is very close to stoichiometric. We know the intent of the burner commissioning process is to set it to be near stoichiometric at maximum fire, so we will use this assumption.

For a given mass flow rate of natural gas to the burner, $\dot{m}_{NG}$, the maximum rate of heat generation, $\dot{Q}_{burner}$, is given by

$$\dot{Q}_{burner} = \dot{m}_{NG}\left(ngP.LM\right) \qquad \text{bsc.1}$$

Given the definition of lower heating value, we assume the energy generated by the combustion of air and natural gas is used to heat the burner exhaust to the adiabatic flame temperature from the energy level it would have if the combustion products were at the initial ambient condition. If the enthalpies of the burned gas mixture are given by h_{amb} and h_{adia} for the ambient and maximum flame temperature conditions respectively, we can write

$$\dot{Q}_{burner} = \dot{m}_{burner}\left(h_{adia} - h_{amb}\right) \qquad \text{bsc.2}$$

where $\dot{m}_{burner}$ is the total mass flow of air and natural gas through the burner. Based on our assumption of stoichiometric combustion, we can write

$$\dot{m}_{burner} = \dot{m}_{NG} + \dot{m}_{air} = \dot{m}_{NG}\left(1 + ngP.RW\right) \qquad \text{bsc.3}$$

Mixing Heat Transfer

Heat transfer to the engine exhaust stream will occur through mixing. But since no chemical reactions are involved, we can treat the two streams as though they are separate. The amount of heat transferred to the exhaust stream, $\dot{Q}_{trans}$, to achieve a target SCR temperature and corresponding enthalpy, $h_{scr,burner}$, is given by

$$\dot{Q}_{trans} = \dot{m}_{burner}\left(h_{adia} - h_{scr,burner}\right) \qquad \text{bsc.4}$$

The goal is to use this energy to raise the temperature of the exhaust stream when it hits the SCR bed from the current measured value and associated enthalpy, $h_{meas,exh}$, to the target temperature and enthalpy. We can therefore write

$$\dot{Q}_{trans} = \dot{m}_{exh}\left(h_{scr,exh} - h_{meas,exh}\right) \qquad \text{bsc.5}$$

Note that there is a distinction between the enthalpy of the two streams at the desired SCR bed temperature – the temperature is the same but the stream composition is different.

Minimum Natural Gas Flow Rate

We can combine our first three equations to give

$$\begin{aligned} \dot{Q}_{burner} &= \dot{m}_{NG}\left(ngP.LM\right) \\ &= \dot{m}_{burner}\left(h_{adia} - h_{amb}\right) \\ &= \dot{m}_{NG}\left(1 + ngP.RW\right)\left(h_{adia} - h_{amb}\right) \end{aligned} \qquad \text{bsc.6}$$

which we can solve for the enthalpy at the burner adiabatic flame temperature

$$h_{adia} = \frac{\left(ngP.LM\right)}{\left(1 + ngP.RW\right)} + h_{amb} \qquad \text{bsc.7}$$

Setting the heat transfer equations equal gives us

$$\begin{aligned} \dot{Q}_{trans} &= \dot{m}_{exh}\left(h_{scr,exh} - h_{meas,exh}\right) \\ &= \dot{m}_{burner}\left(h_{adia} - h_{scr,burner}\right) \\ &= \dot{m}_{NG}\left(1 + ngP.RW\right)\left(h_{adia} - h_{scr,burner}\right) \end{aligned} \qquad \text{bsc.8}$$

which we can solve for the mass flow rate of natural gas

$$\dot{m}_{NG} = \frac{\dot{m}_{exh}\left(h_{scr,exh} - h_{meas,exh}\right)}{\left(1 + ngP.RW\right)\left(h_{adia} - h_{scr,burner}\right)} \qquad \text{bsc.9}$$

Substituting the equation for the enthalpy of the burner exhaust at the adiabatic flame temperature gives us our final equation for minimum required natural gas burner flow rate

$$\dot{m}_{NG} = \frac{\dot{m}_{exh}\left(h_{scr,exh} - h_{meas,exh}\right)}{\left(ngP.LM\right) - \left(1 + ngP.RW\right)\left(h_{scr,burner} - h_{amb}\right)} \qquad \text{bsc.10}$$

The form of the denominator of this equation provides a hint as to why burners are often improperly sized. The lower heating value represents the amount of energy produced *only if the products of combustion are returned to the pressure and temperature of the reactants*. The second term in the denominator represents the amount of energy that is not available because the temperature of the combustion products is only reduced to the target SCR bed temperature. Increasing either the burner air/fuel ratio or the target temperature for the SCR bed will make the second term larger which will increase the amount of natural gas required.

We should also note that we are making the rosiest possible assumptions about combustor efficiency and heat loss. The heat loss due to the higher temperature of the burner exhaust will be certain to lead to higher heat transfer along the pipe leading to the aftertreatment system. The natural gas flow rate we compute will actually represent the minimum possible required to achieve our target SCR bed temperature.

It is common for natural gas burners to be sized in terms of the “standard” volume flow rate. The volume flow rate is related to the mass flow rate by

$$\dot{V}_{NG} = \frac{\dot{m}_{NG}}{\rho_{NG}} = \dot{m}_{NG}\frac{R_u T}{\left(ngC.MM\right)P} \qquad \text{bsc.11}$$

where R_u is the ideal gas constant, and ρ_{NG} is the natural gas density at the absolute temperature T and absolute pressure P. The use of “standard” volume flow is unfortunate since there are at least three different “standards” in common usage for natural gas. The Maxon flow meter used on the current burner system references a standard pressure of 14.73 psia and standard temperature of 60 F.

Burned Gas Composition and Properties

To solve the equation for required natural gas flow rate, we need to know the enthalpies of both the burner and engine exhaust streams at three different temperatures. The `gas_prop` task requires temperature, pressure and stream composition as inputs. Lacking a good pressure measurement, we will assume atmospheric pressure for all these calculations.

The `burn_emis` task can be used to calculate the equilibrium burned gas composition given the composition of the reactant streams. For the burner, we assume the reactants are natural gas with the composition determined by the ONGA. We assume the air has a moisture content measured by the Vaisala humidity sensor at the inlet to the combustion air handler. We continue to assume stoichiometric combustion.

For the engine exhaust gas combustion, we assume the air has the moisture content measured by the air handler and the diesel fuel has the H/C ratio specified in the composition specification file maintained at the test cell. We use the measured air flow rate from the test cell and back-calculate the fuel rate using chemical balances based on the measured concentrations of carbon bearing species in the exhaust. The appropriate property variables for enthalpy are of the form `streamP.TH where stream` is the name given to either the engine exhaust or burner exhaust stream at the appropriate temperature.

The calculation for natural gas flow rate is specified in the `gen_labels.seec` file at the test cell and is performed in real-time by the `compvar` task. The result is logged continuously whenever the engine is running. If the burner is locked out, the values are easy to interpret. If the burner is active, the result is the excess natural gas flow required above and beyond the natural gas flow recorded by the SEEC software.