San Joaquin Valley
Unified Air Pollution Control District
Best Performance Standard (BPS) x.x.xx

Date: April 17, 2013

Class	*Process Heaters*
Category	*Thermal Fluid Heat Transfer System*
BPS Specification	Thermal heat transfer systems meeting this Best Performance Standard shall comply with all elements listed below: 1. The unit shall be fired with natural gas where natural gas utility service is available. When not available, the unit may be fired on propane, butane or LPG. 2. The thermal fluid heater shall be a forced-draft design. 3. The thermal fluid heater shall be designed to recover heat from the stack sufficient to achieve a stack temperature of no greater than the temperature of the returning heat transfer fluid plus 150 F when operating at design firing rate. 4. The burner and firing controls for the thermal fluid heater shall include an O2 trim control system or other alternate system which is designed to minimize the excess air in the heater exhaust. 5. The combustion air blower shall be powered with a variable speed drive which serves to modulate the flow from the fan to match system demand unless it can be demonstrated to the satisfaction of the APCO that the process requires specific fire safety devices which are incompatible with variable speed drives. 6. The motors driving the combustion air fan and the thermal fluid circulating pump shall be NEMA premium efficiency motors.

Percentage Achieved GHG Emission Reduction Relative to Baseline Emissions	**9.5%**

District Project Number	C-1100388
Evaluating Engineer	Dennis Roberts, P.E.
Lead Engineer	Martin Keast
Public Notice of Intent Date	November 13, 2013
Public Notice: Start Date	3/18/13
Public Notice: End Date	4/17/13
Determination Effective Date	4/17/13

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I. Best Performance Standard (BPS) Determination Introduction

A. Purpose

To assist permit applicants, project proponents, and interested parties in assessing and reducing the impacts of project specific greenhouse gas emissions (GHG) on global climate change from stationary source projects, the San Joaquin Valley Air Pollution Control District (District) has adopted the polic*y: District Policy – Addressing GHG Emission Impacts for Stationary Source Projects Under CEQA When Serving as the Lead Agency.* This policy applies to projects for which the District has discretionary approval authority over the project and the District serves as the lead agency for CEQA purposes. Nonetheless, land use agencies can refer to it as guidance for projects that include stationary sources of emissions. The policy relies on the use of performance based standards, otherwise known as Best Performance Standards (BPS) to assess significance of project specific greenhouse gas emissions on global climate change during the environmental review process, as required by CEQA. Use of BPS is a method of streamlining the CEQA process of determining significance and is not a required emission reduction measure. Projects implementing BPS would be determined to have a less than cumulatively significant impact. Otherwise, demonstration of a 29 percent reduction in GHG emissions, from business-as-usual, is required to determine that a project would have a less than cumulatively significant impact.

B. Definitions

Best Performance Standard for Stationary Source Projects for a specific Class and Category is the most effective, District approved, Achieved-in-Practice means of reducing or limiting GHG emissions from a GHG emissions source, that is also economically feasible per the definition of Achieved-in-Practice. BPS includes equipment type, equipment design, and operational and maintenance practices for the identified service, operation, or emissions unit class and category.

Business-as-Usual is - the emissions for a type of equipment or operation within an identified class and category projected for the year 2020, assuming no change in GHG emissions per unit of activity as established for the baseline period, 2002-2004. To relate BAU to an emissions generating activity, the District proposes to establish emission factors per unit of activity, for each class and category, using the 2002-2004 baseline period as the reference.

Category is - a District approved subdivision within a “class” as identified by unique operational or technical aspects.

Class is - the broadest District approved division of stationary GHG sources based on fundamental type of equipment or industrial classification of the source operation.

C. Determining Project Significance Using BPS

Use of BPS is a method of determining significance of project specific GHG emission impacts using established specifications. BPS is not a required mitigation of project related impacts. Use of BPS would streamline the significance determination process by pre-quantifying the emission reductions that would be achieved by a specific GHG emission reduction measure and pre-approving the use of such a measure to reduce project-related GHG emissions.

GHG emissions can be directly emitted from stationary sources of air pollution requiring operating permits from the District, or they may be emitted indirectly, as a result of increased electrical power usage, for instance. For traditional stationary source projects, BPS includes equipment type, equipment design, and operational and maintenance practices for the identified service, operation, or emissions unit class and category.

II. Summary of BPS Determination Process

The District has established Thermal Fluid Heat Transfer System as a separate class and category which requires implementation of a Best Performance Standard (BPS) pursuant to the District's Climate Change Action Plan (CCAP). The District's determination of the BPS for this class and category has been made using the BPS development process established in the District's Final Staff Report, *Addressing Greenhouse Gas Emissions under the California Environmental Quality Act.* A summary of the specific implementation of the phased BPS development process for this specific determination is as follows:

Table 1 BPS Development Process Phases for Thermal Fluid Heat Transfer System			
Phase	**Description**	**Date**	**Description**
1	Public Notice of Intent	11/13/12	The District's intent notice is attached as Appendix A
2	BPS Development	3/12/13	See evaluation document.
3	Public Participation: Public Notice Start Date	3/18/13	A Draft BPS evaluation was provided for public comment.
4	Public Participation: Public Notice End Date	4/17/13	All public comments received and the District's responses are attached as Appendix C
5	Public Workshop	N/A	No workshop was conducted for this BPS determination.
6	Finalization	4/22/13	The BPS established in this evaluation document will be effective on the date of finalization.

III. Class and Category

Process Heaters are recognized as a distinct class based on the following:

A process heater is defined as any fuel fired combustion equipment which transfers heat from combustion gases to fluid or process streams.

Process heaters are a distinct class with respect to the District's prohibitory rules for criteria pollutant emissions (Rules 4306, 4307, 4308 and 4320).

Process heaters differ from boilers and steam generators in that process heaters are not limited to boiling or raising the temperature of water.

Process heaters differ from dryers as process heaters do not dry or cure material by direct contact with the products of combustion.

Thermal Fluid Heat Transfer Systems typically consist of a thermal fluid heater with piping, pump(s) and other ancilliary equipment in which a liquid phase heat transfer medium is heated and circulated to one or more heat energy users within a closed loop system. Thermal oil, glycol, and water are common heat transfer mediums as well as specialized circulating fluids such as cooking oil which is circulated through food frying systems. Appendix A provides a schematic illustration of a typical system. Thermal fluid heat transfer systems are recognized as a distinct category of the class “process heaters”.

IV Public Notice of Intent

Prior to developing the BPS for this class and category, the District published a Notice of Intent. Public notification of the District’s intent to develop BPS for this class and category was sent on November 13, 2013 to individuals registered with the CCAP list server. The District’s notification is attached as Appendix C.

No comments were received during the initial public outreach.

V. BPS Development

STEP 1. Establish Baseline Emissions Factor for Class and Category

The Baseline Emission Factor (BEF) is defined as the three-year average (2002-2004) of GHG emissions for a particular class and category of equipment in the San Joaquin Valley (SJV), expressed as annual GHG emissions per unit of activity. The Baseline Emission Factor is calculated by first defining an operation which is representative of the average population of units of this type in the SJV during the Baseline Period and then determining the specific emissions per unit throughput for the representative unit.

A. Representative Baseline Operation

For thermal fluid heaters the *r*epresentative baseline operation has been determined to be a 2-pass helical coil thermal fluid heater with a net thermal efficiency of 80% (NHV), fired with natural gas. The baseline unit is forced draft design. The burner and firing controls are a conventional air/fuel ratio control without O2 trim, assumed to operate with a stack O2 concentration of 6% dry basis. The combustion air blower and heat transfer fluid circulation pump are fixed speed equipment driven by conventional-efficiency electric motors. This determination is based on:

- A 2-pass helical coil desigh is historically the typical configuration and the design with lowest 1st cost for modern thermal fluid heaters. The unit typically consists of a single helical heat transfer coil through which the heat transfer fluid flows, mounted in a cylindrical heater casing with the burner firing at one end of the casing, centered in the coil. Combustion occurs in the inner coil space which is the primary zone of radiant heat transfer in the unit (1st pass). Combustion gases flow to the stack by reversing direction and flowing on the outside of the coil (2nd pass), returning to the firing end of the unit where the stack is located. The second pass is the primary zone of convective heat transfer in the unit. The thermal fluid heater in place during 2002-2004 is exprected to have had an average age of 10 years, and therefore was installed in the early 1990's. It is reasonable to assume (and historically consistent with general industrial practice) that the lower 1st cost of a simple 2-pass unit would have out-weighed the advantages of the enhanced energy efficiency available from a 3-pass design or of a unit equipped with an economizer at the time of installation.

- An average thermal efficiency of 80% (LHV) is estimated for the 2-pass design based on manufacturer's literature which suggests thermal efficiencies in the range of 75-85% (LHV) for two-pass thermal fluid heaters as well as on references from the literature which suggest a thermal efficiency in the low 80's as historically representative of process heater performance in general.

- Assumptions concerning firing controls and drivers for combustion air fans and pumps are generally consistent with current standard installation practices which were also in effect at the time time of the baseline period.

B. Basis and Assumptions of Analysis for Baseline Emissions

- All direct GHG emissions are produced due to combustion of natural gas in this unit.
- The higher heating value (HHV) for natural gas is 111% of the lower heating value (LHV).
- GHG emissions are stated as "CO_2 equivalent" (CO_2e) which includes the global warming potential of methane and nitrous oxide emissions associated with gaseous fuel combustion.
- The GHG emission factor for natural gas combustion is 117 lb-CO_2e/MMBtu (HHV) based on published emission factors by the California Air Resources Board
- F-Factor for Natural Gas: 8710 dscf/MMBtu (HHV) at 68°F (40 CFR 60, Appendix B)

- 385.5 = number of cubic feet of ideal gas in a lb-mole at 68 F and atmospheric pressure.
- F-Factor correction for excess air in the stack gas is A_d = 20.8%/(20.8%-%O2)
- The heater is assumed to operate at 6% O2 in the exhaust.
- Indirect emissions produced due to operation of the combustion air fan will and the circulating pump will be considered in the analysis.
- For the baseline, electric motors are assumed to be conventional motors with efficiency of 85%.
- The circulating thermal oil pump is assumed to operate at 62 gpm to deliver 1 MMBtu/hr with a pressure differential of 75 psi and a hydraulic efficiency of 65% (based on typical circulating rates and motor horsepowers listed in manufacturer's litereature).
- The combustion air fan is assumed to operate with a static head of 25" water column at a static efficiency of 60%.
- Brake horsepower required for operation of the combustion air fan will be calculated by the following equation which is a simplified representation of adiabatic compression ignoring the compressibility of air as appropriate for the low pressure differentials typical for fans:

$$\text{Bhp(fan)} = \frac{\text{Air Flow, CFM}}{6{,}356} \times \frac{\text{Static Head, in. of water}}{\text{Static Efficiency}}$$

- Brake horsepower required for operation of the circulating pump will be calculated by the following equation which applies the pump efficiency to the calculated hydraulic horsepower to determine the brake horsepower requirement:

$$\text{Bhp (pump)} = \frac{\text{flow, gpm}}{1{,}713} \times \frac{\text{Differential Pressure, psi}}{\text{Efficiency}}$$

- The combustion air rate is assumed to be essentially equal to the dry stack gas rate in standard cubic feet.
- Indirect emissions from electric power consumption are calculated based on the draft District FYI "Quantifying Greenhouse Gas Emission Due to Electricity Use" which identifies an emission factor of 0.690 lb-CO_2e per kWh.

C. Unit of Activity

To relate Business-as-Usual to an emissions generating activity, it is necessary to establish an emission factor per unit of activity, for the established class and category, using the 2002-2004 baseline period as the reference.

The resulting emissions factor is the combination of
GHG emission reductions achieved through technology, and
GHG emission reductions achieved through changes in activity efficiencies.

A unit of activity for this class and category will be taken as 1,000,000 Btu's of process heat supplied by the circulating thermal heat transfer fluid.

For purposes of development of the GHG emission factors, it will be assumed that GHG emissions reductions achieved through changes in activity efficiencies are not significant. This assumption has been made based on:

- This class and category of equipment is used at a wide range of facilities, diverse in operation and size, making it difficult to characterize specific efficiency improvements.

- A search of available literature did not yield any data which would support a direct estimate of changes in GHG emission from heat transfer systems in this class and category since the baseline period based on changes in activity efficiencies.

D. Calculations

STEP 1. Establish the Baseline Emission Factor per Unit of Activity

The Baseline Emission Factor (BEF) is the sum of the direct (GHG_D) and indirect (GHG_I) emissions (on a per unit of activity basis), stated as lb-CO_2 equivalent:

$BEF = GHG_D + GHG_I$

<u>Direct Emissions:</u>

$GHG_D = E_f \times SFC$

where,

E_f = GHG emission factor = 117 lb- $CO_{2(e)}$/MMBtu of natural gas

SFC = Specific Fuel Consumption = MMBtu fired (HHV) per unit of activity

For a net thermal efficiency (LHV basis) of 80% (Baseline),

SFC = 111% (HHV/NHV) x 1,000,000 MMBtu-supplied ÷ 80% (LHV)
= 1.39 Btu-fired (HHV)/Btu-supplied
= 1.39 MMBtu-fired(HHV) per Unit of Activity

Direct emissions are then calculated as:

GHG_D = 117 lb-$CO_{2(e)}$/MMBtu (HHV) x 1.0 MMBtu/hr supplied x 1.39
= 162.6 lb-CO2(e)/MMBtu-supplied

Indirect Emissions

Only indirect emissions associated with the operation of electric motors are considered determined by the following:

GHG (electric motor) = Electric Utility GHG Emission Factor x kWh consumed

Thermal Fluid Circulating Pump

$$\text{Bhp} = \frac{62\text{ gpm} \times 75\text{ psi}}{1{,}713 \times 65\%}$$

$$\text{Bhp} = 4.18$$

Combustion Air Fan

$$\text{Combustion Air Rate} = 1.39\ \text{MMBtu/hr} \times 8710\ \text{scf/MMBtu} \times \frac{20.85\%}{20.85\% - 6.0\%} \times \frac{1\text{ hour}}{60\text{ minutes}}$$

$$\text{Combustion Air Rate} = 283\text{ cfm}$$

$$\text{Bhp} = \frac{283\text{ CFM} \times 25''\text{ W.C.}}{6{,}356 \times 60\%}$$

$$\text{Bhp} = 1.86$$

Total bhp for both motors is then converted to a brake kWh per MMBtu:

Brake kWh: = (4.18 + 1.86) x 0.7457 kWh/bhp = 6.04 kWh/MMBtu

Actual kWh consumption is calculated based on motor efficiency:

kWh consumption = 6.04 brake kWh/MMBtu ÷ 85% = 7.10 kWh/MMBtu supplied

GHG (electric motor) = 0.69 lb-CO_2e/kWh x 7.10 kWh/MMBtu

GHG (electric motor) = 4.9 lb-CO_2e/MMBtu

The Baseline Emission Factor is the sum of the direct and the indirect emissions:

BEF = *162.6 + 4.9 = 167.5 lb-* CO_2(e)*/MMBtu supplied*

STEP 2. List Technologically Feasible GHG Emission Reduction Measures

<u>Fire the unit with only natural gas</u>

Where and when available, the use of natural gas provides the lowest GHG emission rate per Btu

<u>Enhanced recovery of heat from the stack gas</u>

Installing additional convective heat transfer surface in the unit allows enhanced recovery of heat from the stack gases. The resulting improvement in thermal efficiency results in reduced GHG emissions per MMBtu of delivered process heat. Increased convective heat transfer is achieved by increasing the internal convective surface (such as by specifying a 3-pass heater) and/or installing an external convection section. An external convection section (also known as an economizer) is used to preheat the thermal fluid before entering the heater proper, recovering heat from the stack gas. Use of an external convection section may also allow recovery of stack heat for services other than preheating the thermal oil, such as other low level process heat demands, HVAC, etc., when present at the facility.

Use of an air preheater, which transfers heat from the stack to the combustion air using an external heat exchchanger and retuning the heat to the unit, may also be used and it improves thermal efficiency in a manner similar to increased convective surface. However, the use may be limited since increased air preheat may increase NOx emissions.

Minimize Excess Air

A. Forced Draft Design

The combustion process generally requires an excess of air (air in excess of the stochiometric requirement for combustion of the fuel) to ensure efficient combustion and safe operation. Operations which exceed the minimum amount of excess air required for clean and safe operation result in a loss of efficiency as a result of the increased stack losses. Higher levels of excess air are especially characteristic of natural draft units due to 1) reduced capability to safely control the combustion air rate, especially in turndown conditions, and to 2) air inleakage into the unit caused by the negative pressure present in the unit. Specification of a forced draft unit helps reduce excess air in the process.

B. Limit excess air and/or Flue Gas Recirculation (FGR)

The excess air and FGR requirement for any particular unit will be affected substantially by the allowed level of NOx emissions and by the selected NOx emission control strategy. Units installed in the San Joaquin Valley are generally required to meet stringent air NOx emission limits and thus can be expected to be equipped with the most advanced emission control technology. Units operating with ultra low NOx burners without flue gas recircultion may rely on high excess air as a diluent to reduce peak flame temperature. Burners with flue gas recirculation (FGR) may require somewhat lower levels of excess air but produce substantial indirect GHG emissions due to operation of the FGR fan. Use of ultra low NOx burners both with and without FGR generally results in increased indirect GHG emissions relative to conventional burners. However, units equipped with selective catalytic reduction (SCR) for NOx control offer operation with the lowest achievable level of excess air since these units may operate with conventional burners with no or only minimal FGR and may be specified for operation with O2 levels at or below 3%.

C. O2 Trim Control Instrumentation

When burners are manually tuned on a periodic basis, they are typically adjusted to a conservatively high excess air value, ensuring safe operation over the entire operating range of the boiler but negatively impacting the average thermal efficiency of the unit.

O2 trim instrumentation serves to continuously monitor the O2 concentration in the stack and to continuously trim the air/fuel ratio in the combustion chamber to the minimum value required for stable combustion and for control of NOx emissions if applicable, over the entire operating range of the unit. Other control systems (such as pre-programed ratio control systems which provide a repeatable optimum air/fuel ratio over the full operating range of the unit) are also available which serve to minimize excess air and which may better fit the specific process. Such systems are considered equivalent to O2 trim control.

Use of premium efficiency motors

An electric motor efficiency standard is published by the National Electrical Manufacturers Association (NEMA) which is identified as the "NEMA Premium Efficiency Electric Motors Program". For large motors, the NEMA premium efficiency motor provides a gain of approximately 5-8 percentage points in motor efficiency when compared to a standard efficiency motor. The NEMA specification covers motors up to 500 horsepower and motors meeting this specification are in common use and are available from most major electric motor manufacturers.

Use of speed control for the combustion air

Control of the operation of the combustion air fan by use of a variable speed electric motor will provide substantial energy savings when compared to operation at a fixed speed and controlled by throttling the discharge flow. The most common and economical variable speed drive is the variable frequency drive (VFD) which has become commonly available in recent years and is typical for new combustion air fan applications. For the combustion air fan, the VFD provides especially significant energy savings when a unit is operated at substantial turndown ratios which can result in throttling away more than half the rated energy output of the motor. In some specialized cases, a speed controlled fan may not be compatible with the heater's required flame safety system.

Use of speed control for the heat transfer fluid circulation pump

Control of the operation of the heat transfer fluid circulation pump by use of a variable speed electric motor could also provide substantial energy savings when compared to operation at a fixed speed and controlled by throttling the discharge flow. Typical operation of the heat transfer fluid circulation is depicted by the schematic in Appendix A. As shown, the circulating fluid loop operates through a back pressure control valve while the pump continues to operate at a fixed speed and horsepower input. At low fluid utilization rates by the process, substantial energy is wasted in this mode. A variable speed pump would match the pump output to the actual demand down to the lower limit of flow for the heater, eliminating the backpressure throttling operation.

Use of High Efficiency Combustion Air Fans and Thermal Oil Circulating Pumps

The peak efficiency of centrifugal fans and pumps may vary from 60 to 80% depending upon design and application. Use of a higher efficiency fan or pump provides savings in indirect GHG emissions due to the significant reduction in electric motor horsepower required.

Table 2 Technologically Feasible GHG Control Measures for *Thermal Fluid Heat Transfer Systems*	
GHG Control Measures	**Qualifications**
Natural gas firing	Reduces heat loss to the stack, significantly increasing thermal efficiency
Forced draft design	Reduces excess air in stack gas, increasing thermal efficiency
Enhanced recovery of stack heat	Reduces heat loss to the stack, significantly increasing thermal efficiency
Use O2 trim or equivalent	Reduces excess air, particularly for turn down operations, increasing thermal efficiency
Limit excess air	Reduces excess air, increasing thermal efficiency but may require installation of SCR due to SJVAPCD NOx regulations
Variable speed fans and pumps	Reduces electrical power requirement for operation of the thermal heat transfer system resulting in reduction of indirect GHG emissions
Premium efficiency motors	Reduces electrical power requirement for operation of the thermal heat transfer system resulting in reduction of indirect GHG emissions
Use of High Efficiency Combustion Air Fans and Thermal Oil Circulating Pumps	Reduces electrical power requirement for operation of the thermal heat transfer system resulting in reduction of indirect GHG emissions

All of the control measures identified above operate in conjunction with control equipment for criteria pollutants which meets current regulatory requirements. None of the identified control measures would result in an increase in emissions of criteria pollutants.

STEP 3. Identify all Achieved-in-Practice GHG Emission Control Measures

For all technologically feasible GHG emission reduction measures, all GHG reduction measures determined to be Achieved-in-Practice are identified. Achieved-in-Practice is defined as any equipment, technology, practice or operation available in the United States that has been installed and operated or used at a commercial or stationary source site for a reasonable period of time sufficient to demonstrate that the equipment, the technology, the practice or the operation is reliable when operated in a manner that is typical for the process. In determining whether equipment, technology, practice or operation is Achieved-in-Practice, the District will consider the extent to which grants, incentives or other financial subsidies influence the economic feasibility of its use.

The following findings or considerations are applicable to this class and category:

Natural gas firing:

All thermal fluid heat transfer systems currently permitted by the District are fired with natural gas. Therefore natural gas firing is achieved-in-practice.

Forced Draft Design

Forced draft design is representative of the current commercial offerings by essential all manufacturers of thermal heat transfer fluid systems and is commonly known to be in widespread commercial operation. Forced draft design is therefore considered achieved-in-practice.

Enhanced Recovery of Heat from the Stack Gases

Thermal fluid heaters featuring 3-pass designs, air preheaters and/or external convection sections are in commercial operation and are standard options routinely offered by thermal fluid heater manufacturers. A specified minimum level of performance based on the use of extended convective heat transfer surface is thus required to represent achieved-in-practice technology. The following considerations are applicable to the achieved-in-practice status of this technology:

- Quoted thermal efficiencies of up to 93.0%, lower heating value (LHV) basis, are available from thermal fluid heater manufacturers based upon use of external convection sections. For purposes of this analysis, it is conservatively assumed that such efficiency would only be achieved at a low heat transfer fluid supply temperature (outlet of the heater) for the circulating heat transfer fluid ($\leq$250 F).
- Review of the literature[1,2] pertaining to high efficiency design of fired process heaters indicates that a temperature approach (stack temperature – feed temperature) of 100-200 F is achieved-in-practice.
- The District estimates that a temperature approach of 150 F would be representative of a design with sufficient convective surface area to achieve

[1] Garg, A., "How to Boost the Performance of Fired Heaters", Chemical Engineering, November, 1989.
[2] Garg, A., "Revamp Fired Heaters to Increase Capacity", Hydrocarbon Prcessing, June, 1998.

a thermal efficiency of approximately 90% (LHV) when operating with a supply temperature of 250 F for the circulating heat transfer fluid (see Appendix B). Therefore, specification of a temperature approach of 150 F (stack temperature – heat transfer fluid return temperature) at rated firing capacity is consistent with published manufacturer's information and is determined to represent the achieved-in-practice application of this technology for thermal heat transfer systems.

Limit excess air and/or Flue Gas Recirculation (FGR)

Specification of percentage O2 in the stack or percentage FGR is problematic for thermal heat transfer systems since these units primarily control NOx emissions using ultra-low NOx burners and FGR and these parameters may vary widely depending upon the selected burner technology. A restrictive specification in this regard would likely dictate installation of an SCR system for NOx control for units meeting the District's emission regulations for NOx, given the current state of NOx control technology. However, in the case of thermal fluid heaters, the District does not currently consider the use of SCR to be an Achieved-in-Practice technology for thermal fluid heaters and therefore the NOx control strategy for thermal fluid heaters is assumed be based only on burner technology for purposes of the BPS determination. Based on this, placing limits on excess air and flue gas recirculation rates is not considered to be a feasible GHG reduction measure for thermal fluid heaters.

O2 Trim Control Instrumentation

O2 Trim instrumentation or equivalent is commercially available and is commonly known to be in widespread commercial use for fired equipment in general. O2 trim instrumentation is therefore considered achieved-in-practice.

Use of high efficiency pumps and fans

The absolute value of efficiency which can be achieved by a fan or pump is highly dependent upon the specific operating conditions including flow, pressure, and temperature, all of which may vary significantly for any specific application. Given this variability as well as the absence of any effective industry standard for fan or pump efficiency, the District's opinion is that the specification of fan or pump efficiency cannot be realistically included as a technologically feasible reduction at this time. Therefore, the use of high efficiency pumps and fans is not considered achieved-in-practice.

Use of premium efficiency motors

Premium efficiency motors and motor speed control (variable frequency drive) are commercially available and are commonly known to be in widespread commercial use for fired equipment in general. Premium efficiency motors and motor speed control are therefore considered achieved-in-practice.

Use of motor speed control

Motor speed controls (variable frequency drives) are commercially available and are commonly known to be in widespread commercial use for combustion air fans on fired equipment in general. Application of this technology to the heat transfer circulating pumps, although technically feasible, presents some specific technical challenges with respect to control system design due to requirements to maintain a minimum flow rate to the heater independent of the process demand plus other control considerations related to multiple process users on a single heat transfer supply loop. The District has not identified any operations currently using this technology for the heat transfer fluid circulating pump. Therefore, motor speed control is therefore considered achieved-in-practice for combustion air fans only. Application to the heat transfer fluid circulation pump is not currently considered to be achieved in practice.

Table 3 **Achieved-in-Practice GHG Control Measures for** ***Thermal Heat Transfer Systems***	
GHG Reduction Measures	**Achieved-Qualifications**
The unit shall be fired with natural gas where natural gas utility service is available. When not available, the unit may be fired on propane, butane or LPG.	GHG emissions from natual gas combustion are the lowest of any commecially available fuel.
The thermal fluid heater shall be a forced-draft design	Forced draft design is representative of the current commercial offerings of essentially all manufacturers of thermal heat transfer fluid systems and is commonly known to be in widespread commercial operation.
The thermal fluid heater shall be designed with sufficient convective heat transfer surface to achieve a stack temperature which is no greater than 150 F higher than the design return tempertaure of the heat transfer fluid when operating at maximum firing rate.	Units with extended convective heat transfer surface (relative to a 2-pass unit) are in common use. The District has determined that a temperature approach of 150 F represents the current level for achieved-in-practice application of this technology.
The firing controls for the thermal fluid heater shall incorporate an O2 trim system or equivalent designed minimize excees air in the exhaust of the unit.	O2 Trim instrumentation is commercially available and is commonly known to be in widespread commercial use for fired equipment in general.
The combustion air blower shall be powered with a variable speed drive which serves to modulate the flow from the fan to match system demand unless the process specific flame safety system is not compatible.	Motor speed control (variable frequency drive) is commercially available and is commonly known to be in widespread commercial use for fired equipment in general.
The motors driving the combustion air fan and the thermal fluid circulating pump shall be NEMA premium efficiency motors.	Premium efficiency motors and Motor speed control (variable frequency drive) are commercially available and are commonly known to be in widespread commercial use for fired equipment in general.

STEP 4. Quantify the Potential GHG Emission and Percent Reduction for Each Identified Achieved-in-Practice GHG Emission Control Measure

For each Achieved-in-Practice GHG emission reduction measure identified:
a. Quantify the potential GHG emissions per unit of activity (G_a)
b. Express the potential GHG emission reduction as a percent (G_p) of Baseline GHG emissions factor per unit of activity (BEF)

All Achieved-in-Practice reduction measures are independently implemented and are thus additive in impact. Therefore, the GHG emission quantification will be presented as a single value based on the additive contribution of each individual measure incorporated into the overall control measure.

A. Additional Basis and Assumptions Applicable to the BPS Case:

- *The thermal fluid heater is designed to operate with a temperature approach of 150 F at rated firing capacity.*
- *The unit is assumed to operate with an average O2 concentration in the exhaust of 4.5% due to operation of the O2 trim control.*
- *Hot oil supply temperature is assumed to be 350 F. (typical temperature level for utility steam in industrial facilities).*
- *The following are typical properties for heat transfer fluids as a basis for this calculation:*

Heat Transfer Fluid Properties at Operating Condition	
Density lb/gallon	6.4
Heat Capacity Btu/lb-F	0.58

- *At the given heat transfer fluid circulation rate of 62.5 gallons per minute:*

$$\text{Circulation Rate in lb/hr} = M = 62.5 \text{ gpm} \times 60 \frac{\text{min}}{\text{hour}} \times 6.4 \frac{\text{lb}}{\text{gallon}}$$

M = 24,000 lb/hr

$$\text{Oil Temperature} = \Delta T = \frac{Q}{\quad} \quad \underline{\text{Btu}}$$

$$\text{Increase in Heater F} \quad \frac{\text{hour}}{M \ \frac{\text{lb}}{\text{hour}} \times C_p \ \frac{\text{Btu}}{\text{lb-F}}}$$

$$\Delta T = \frac{1{,}000{,}000 \ \frac{\text{Btu}}{\text{hour}}}{24{,}000 \ \frac{\text{lb}}{\text{hour}} \times 0.58 \ \frac{\text{Btu}}{\text{lb-F}}}$$

$$\Delta T = 72 \text{ F}$$

$$\text{Oil Return Temperature F} = \text{Supply Temperature} - \Delta T$$

$$\text{Oil Return Temperature F} = 350 - 72 = 278 \text{ F}$$

$$\text{Stack Temperature} = \text{Oil Return Temperature} + \text{Approach}$$

$$\text{Stack Temperature} = 278 \text{ F} + 150 \text{ F} = 428 \text{ F}$$

- *Enthalpy values are referenced to products of combustion at 68 F, water in the vapor state.*

- *The following definitions are applicable for purposes of this analysis:*

  Flue Gas = stoichiometric products of combustion of natural gas, dry basis

  Combustion Water = water vapor produced by combustion of natural gas

  Excess air = additional air provided to the combustion process as required for operation at the specified oxygen concentration at the boiler exhaust.

  Stack Gas = combined boiler exhaust including flue gas, combustion water and excess air.

- *Electric motors powering the combustion air fan and circulating pump are NEMA Premium Efficiency with rated efficiency of 92%.*

- *An energy savings of 30% for electric motor operation associated with the combustion air fan is assumed for units equipped with variable speed drives.*

- The following gas phase properties are applicable based on combustion of methane::

Gas Phase Properties Based on Methane Combustion		
	Average Molecular Weight lb per lb-mole	Average Gas Phase Specific Heat Btu/lb-F (68 - 400 F)
Flue Gas	27.6	0.244
Air	28.8	0.248
Combustion Water	18.0	0.450

- Convective and radiation losses are assumed to be 2.5% of the fired duty (HHV) of the oven

- Combustion water rate based on the combustion of methane is
  Fw = 2,018 scf-water vapor per MMBtu (HHV) at 68 F.

- Given the average molecular weight of 27.6 stated in the table above, the lb of stoichiometric flue gas is given by:

  Lb flue gas = SFC x 8,710 scf/MMBtu x 29.9 lb per mol/385.5 scf/mol
  = 675.6 x SFC

- At 4.5% O2, the excess air correction factor is:

  A_c = (20.85%)/ (20.85% - 4.5%)
  = 1.275

- The lb of excess air in the stack gas are a function of the amount of fuel fired and the excess air correction factor:

  Lb excess air = SFC x 8,710 x (A_c – 1) x 28.8 lb per mol/385.5 scf/mol
  = 178.9 x SFC

- The lb of combustion water in the stack gas are a function of the amount of fuel fired and the excess air correction factor:

Lb excess air $= SFC \times F_w \times 18.0/385.5$ scf/mol
$= SFC \times 2{,}018 \times 18.0/385.5$
$= 94.2 \times SFC$

B. Calculation of Potential GHG Emissions per Unit of Activity (G_a):

Direct Emissions

Thermal Fluid Heater Energy Balance:

Heat Inputs	
Fuel Firing Btu/hour (LHV)	= (SFC/111%) x 1,000,000 Btu supplied
Heat Outputs	
Flue Gas Enthalpy from Combustion Btu/hr	= 675.6 lb/MMBtu x SFC x 0.245 Btu/lb-F x (428-68) F
Excess air enthalpy in flue gas Btu/hr	= 178.9 x SFC x 0.248 Btu/lb-F x (428-68) F
Combustion moisture enthalpy Btu/hr	= 94.2 x SFC x 0.450 Btu/lb-F x (428-68) F
Radiation and convection Loss Btu/hr	= 2.5% x SFC x 1,000,000
Supplied Process Heat	= 1,000,000 Btu

Setting Heat Input = Heat Output and solving for the SFC yields:

SFC_{BPS} = 1.27 MMBtu Fuel (HHV)/MMBtu-supplied

Direct emissions are then calculated as:

GHG_D = 117 lb-$CO_{2(e)}$/MMBtu x 1.27 MMBtu/ton chips
= 148.6 lb-CO2(e)/MMBtu-supplied

Indirect Emissions

Only indirect emissions associated with the operation of electric motors are considered determined by the following:
GHG (electric motor) = Electric Utility GHG Emission Factor x kWh consumed

Thermal Fluid Circulating Pump

$$\text{Bhp} = \frac{62 \text{ gpm} \times 75 \text{ psi}}{1{,}713 \times 65\%}$$

$$\text{Bhp} = 4.18$$

Combustion Air Fan

$$\text{Combustion Air Rate} = 1.27 \text{ MMBtu/hr} \times 8{,}710 \text{ scf/MMBtu} \times \frac{20.85\%}{20.85\% - 4.5\%} \times \frac{1 \text{ hour}}{60 \text{ minutes}}$$

$$\text{Combustion Air Rate} = 235 \text{ cfm}$$

$$\text{Bhp} = \frac{235 \text{ CFM} \times 25'' \text{ W.C.}}{6{,}356 \times 60\%}$$

$$\text{Bhp} = 1.54$$

Since the combustion air fan is equipped with avariable speed drive, a savings of 30% in energy consumption is incorporated:

Adjusted Fan Brake Horsepower = Brake HP x (1-30%) = 1.54 x (1-30%)

= 1.08

Total Brake Horsepower for fan and pump:

$BHP_T = 4.18 + 1.08 = 5.26$

Bhp for both motors is then converted to a brake kWh:

Brake kWh: = 5.26 x 0.7457 kWh/bhp = 3.92 kWh/MMBtu

Actual kWh consumption is calculated based on motor efficiency:

kWh consumption = 3.92 brake kWh/MMBtu ÷ 92% = 4.27 kWh/MMBtu supplied

GHG (electric motor) = 0.69 lb-CO_2e/kWh x 4.27 kWh/MMBtu

GHG (electric motor) = 2.95 lb-CO_2e/MMBtu

The Potential GHG emissions per unit of activity (G_a) is the sum of the direct and the indirect emissions:

G_a = 148.6 + 3.0 = 151.6 lb- CO_2(e)/MMBtu supplied

C. Calculation of Potential GHG Emission Reduction as a Percentage of the Baseline Emission Factor (G_p):

G_p = (BEF - G_a) / BEF = (167.5– 151.6)/167.5 = 9.5%

STEP 5. Rank all Achieved-in-Practice GHG emission reduction measures by order of % GHG emissions reduction

Since all listed reduction measures are independent, no ranking is necessary.

STEP 6. Establish the Best Performance Standard (BPS) for this Class and Category

For Stationary Source Projects for which the District must issue permits, Best Performance Standard is – “For a specific Class and Category, the most effective, District approved, Achieved-In-Practice means of reducing or limiting GHG emissions from a GHG emissions source, that is also economically feasible per the definition of achieved-in-practice. BPS includes equipment type, equipment design, and operational and maintenance practices for the identified service, operation, or emissions unit class and category”.

Based on the definition above and the ranking of evaluated technologies, Best Performance Standard (BPS) for this class and category is determined as:

Best Performance Standard for Thermal Fluid Heat Transfer Systems

Thermal heat transfer systems meeting this Best Performance Standard shall comply with all elements listed below:

1. **The unit shall be fired with natural gas where natural gas utility service is available. When not available, the unit may be fired on propane, butane or LPG.**
2. **The thermal fluid heater shall be a forced-draft design.**
3. **The thermal fluid heater shall be designed to recover heat from the stack sufficient to achieve a stack temperature of no greater than the temperature of the returning heat transfer fluid plus 150 F when operating at design firing rate.**
4. **The burner and firing controls for the thermal fluid heater shall include an O2 trim control system or alternate system designed to minimize the excess air in the heater exhaust.**
5. **The combustion air blower shall be powered with a variable speed drive which serves to modulate the flow from the fan to match system demand unless it can be demonstrated to the satisfaction of the APCO that the process requires specific fire safety devices which are incompatible with variable speed drives.**
6. **The motors driving the combustion air fan and the thermal fluid circulating pump shall be NEMA premium efficiency motors.**

STEP 7. Eliminate All Other Achieved-in-Practice Options from Consideration as Best Performance Standard

The following Achieved-in-Practice GHG control measures identified and ranked in the table above are eliminated from consideration as Best Performance Standard since they have GHG control efficiencies which are less than that of the selected Best Performance Standard as stated in Step 6 of this evaluation:

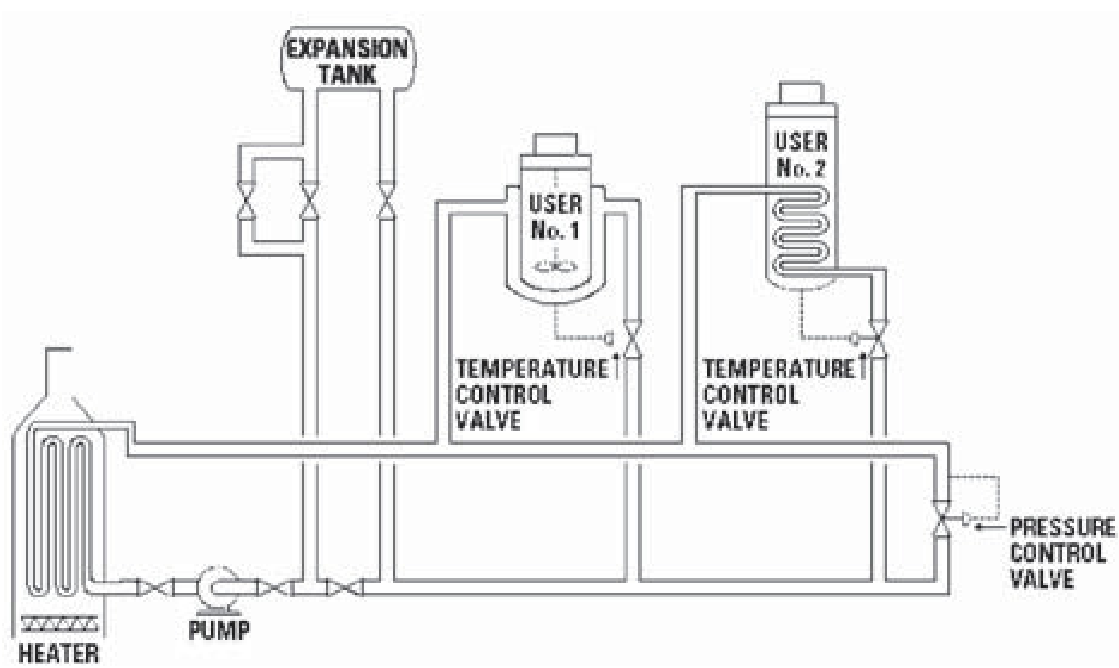
No other Achieved-in-Practice options were identified.

VIII. Appendices

Appendix A: Schematic of typical thermal heat transfer system
Appendix B: Thermal Efficiency Demonstration
Appendix C: Public Notice of Intent: Notice
Appendix D: Public Participation: Notice
Appendix E: Comments Received During the Public Participation Process and Responses to Comments

Appendix A
Typical Thermal Fluid Heat Transfer System

Typical Thermal Fluid Heat Transfer System



Appendix B
Thermal Efficiency Demonstration

90% Net Thermal Efficiency Demonstration for Thermal Fluid Heat Transfer System with BPS

Published manufacturer's information indicates that thermal fluid heat transfer units are offered with net thermal efficiency exceeding 90%. This is conservatively assumed to be applicable to lower temperature systems where the fluid return temperature is low enough to allow substantial recovery of heat from the stack gases. The following calculation demonstrates that the District's GHG reduction measure, requiring that the stack temperature be no greater than 150 F above the fluid return temperature, is consistent with manufacturer's published efficiency value of 90% when evaluated for a low temperature system.

Basis and Assumptions:

- The thermal fluid heater is designed to operate with a temperature approach of 150 F at rated firing capacity.
- Hot oil supply temperature is assumed to be 250 F (lower temperature consistent with use of low pressure steam typical for industrial facilities)
- Unless stated otherwisw, all basis and assumption used in Step 4 for the BPS analysis are applicable.
- At the given heat transfer circulation rate of 62.5 gallons per minute*:*

$$\text{Circulation Rate in lb/hr} = M = 62.5 \text{ gpm} \times 60 \frac{\text{min}}{\text{hour}} \times 6.4 \frac{\text{lb}}{\text{gallon}}$$

$$M = 24{,}000 \text{ lb/hr}$$

$$\text{Oil Temperature Increase in Heater F} = \Delta T = \frac{Q \ \frac{\text{Btu}}{\text{hour}}}{M \ \frac{\text{lb}}{\text{hour}} \times C_p \ \frac{\text{Btu}}{\text{lb-F}}}$$

$$\Delta T = \frac{1{,}000{,}000 \ \frac{\text{Btu}}{\text{hour}}}{24{,}000 \ \frac{\text{lb}}{\text{hour}} \times 0.58 \ \frac{\text{Btu}}{\text{lb-F}}}$$

$$\Delta T = 72 \text{ F}$$

$$\text{Oil Return Temperature F} = \text{Supply Temperature} - \Delta T$$

Oil Return Temperature F	=	250 - 72	=	178 F	
Stack Temperature	=	Oil Return Temperature	+	Approach	
Stack Temperature	=	178 F	+	150 F	= 328 F

Thermal Fluid Heater Energy Balance:

Heat Inputs	
Fuel Firing Btu/hour (LHV)	= (SFC/111%) x 1,000,000 Btu supplied
Heat Outputs	
Flue Gas Enthalpy from Combustion Btu/hr	= 675.6 lb/MMBtu x SFC x 0.244 Btu/lb-F x (328-68) F
Excess air enthalpy in flue gas Btu/hr	= 178.9 x SFC x 0.248 Btu/lb-F x (328-68) F
Combustion moisture enthalpy Btu/hr	= 94.2 x SFC x 0.450 Btu/lb-F x (328-68) F
Radiation and convection Loss Btu/hr	= 2.5% x SFC x 1,000,000
Supplied Process Heat	= 1,000,000 Btu

Setting Heat Input = Heat Output and solving for the SFC yields:

SFC = 1.23 MMBtu Fuel (HHV)/MMBtu-supplied

Converting to an LHV basis:

SFC (LHV) = SFC ÷ 111% = 1.11 MMBtu Fuel (LHV)/MMBtu-supplied

Thermal efficincy is the inverse of the Specific Fuel Consumption (SFC). Therefore,

Thermal efficiency (LHV) = 90%

Appendix C
Public Notice of Intent

San Joaquin Valley
AIR POLLUTION CONTROL DISTRICT

HEALTHY AIR LIVING

NOTICE OF UPDATE of

Best Performance Standards (BPS) for Greenhouse Gas Emissions

NOTICE IS HEREBY GIVEN that the San Joaquin Valley Air Pollution Control District solicits public comment on the updating of Best Performance Standards for the following Stationary Source class and category of greenhouse gas emissions:

Thermal Fluid Heaters

The District's policy for addressing GHG emissions under the California Environmental Quality Act requires that BPS be periodically re-evaluated for incorporation of newly identified GHG emission reduction measures, if available. The District is performing a re-evaluation of the existing Best Performance Standards (BPS) for the subject class and category of greenhouse gas emission source and is soliciting public input on the following topics::

- Recommendations regarding technologies to be evaluated by the District, when re-evaluating Best Performance Standards for the subject Class and Category. Based on a preliminary screening, the District is currently considering the following GHG emission control specifications as applicable to this class and category::
  - Minimum requirements for convection section performance relative to feed temperature
  - Forced draft design required
  - O2 trim control required, maximum O2 concentration specification
  - High efficiency motors with VFD for fans required

The Distict's existing BPS for this class and category can be viewed at http://www.valleyair.org/Programs/CCAP/bps/BPS_idx.htm .

Information regarding the District's Best Performance Standard program can be obtained from the District's website at http://www.valleyair.org/Programs/CCAP/CCAP_idx.htm.

Written comments regarding the proposed Best Performance Standard should be addressed to Dennis Roberts by email, dennis.roberts@valleyair.org, or by mail at SJVUAPCD, 1990 East Gettysburg Ave., Fresno, CA 93726 and must be received by December 14, 2012. For additional information, please contact Dennis Roberts at dennis.roberts@valleyair.org or by phone at (559) 230-5919.

Information regarding the District's Climate Action Plan and how to address GHG emissions impacts under CEQA, can be obtained from the District's website by clicking on http://www.valleyair.org/Programs/CCAP/CCAP_idx.htm.

November 13, 2010

Appendix D
Public Participation: Notice

Dennis Roberts

From: process_heaters_bps@lists.valleyair.org
Sent: Monday, March 18, 2013 2:36 PM
To: David McDonough
Subject: SJVAPCD - Development of Best Performance Standards
Attachments: ATT00001.txt

The San Joaquin Valley Air Pollution Control District is soliciting public comment on the development of Best Performance Standards (BPS). This email is to advise you the draft evaluation for Thermal Fluid Heat Transfer System (Re-evaluation) is now available for review here.

Written comments should be addressed to Dennis Roberts by email, dennis.roberts@valleyair.org, or by mail at SJVUAPCD, 1990 East Gettysburg Avenue, Fresno, CA 93726 and must be received by **April 17, 2013**. For additional information, please contact Dennis Roberts by e-mail or by phone at (559) 230-5919.

Appendix E
Comments Received During the Public Participation Process and Responses to Comments

Comments Received During the Public Participation Process and District Responses to Comments

Stakeholders Written Comments:

Insight Environmental

1. **Comment:** Certain burner and burner control systems may be incompatible with the requirement for a variable speed drive on the combustion air fan. As an example, a heater which provides a recirculating stream of heated cooking oil to a frying process has a requirement to modulate the burner in an on-to pilot control mode with rapid restart in the event of a periodic loss of product feed to the frying system. The required fire safety system (Fireye) does not have the logic to support operation of the variable speed drive. The following modification is suggested with respect to the requirement for the variable speed drive:

    The combustion air blower shall be powered with a variable speed drive which serves to modulate the flow from the fan to match system demand, unless it can be demonstrated to the satisfaction of the APCO that fire safety devices are incompatible with variable speed drives.

District Response: The District concurs. Considering the wide range in types of processes which may be serviced by a thermal fluid heater and the potential burner control and flame safety issues which might arise, the BPS will be revised to incorporate the suggested revision.

2. **Comment:** For many industries O2 trim control for burners is an acceptable way to modulate the fuel/air mixture to a preset exhaust limit. This is usually in the 2 to 3% O2 range. Our experience with O2 sensors in the food industry is that they are problematic and not always reliable. A more suitable system for the food industry to create a more reliable burner air/gas ratio control now uses individual gas and air modulating control valves. The two valves, gas and air are pre-programmed with at least 10 steps through the full burner firing range. At each step, the fuel / air ratio is determined by monitoring the exhaust O2 and the valves are programmed to remember these positions. This system will repeat the fuel air curve from low fire to high fire every time. The requirement for O2 trim should be modified to allow alternative systems which still serve to minimize excess air in the stack gas.

District Response: The District concurs. An allowance for alternate systems which better fit the specific process while still achieving the objective of minimizing the excess air will be included in the BPS.