High Efficiency, Ultra-Low Emission, Integrated Process Heater System

Final Technical Report

Prepared by

Recipient:

TIAX LLC
15 Acorn Park
Cambridge, MA 02140
Project Director: Howard Mason
Tel: 408 517-1550
Fax: 408 517-1551

Project Team Members:

ExxonMobil Research and Engineering Company
3225 Gallows Road
Fairfax, Virginia 22037-0001

Callidus Technologies, LLC
7130 South Lewis, Suite 335
Tulsa, OK 74136

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U. S. Department of Energy
Golden Field Office
1617 Cole Blvd.
Golden, CO 80401
Project Manager: William Prymak
Tel: 303 275-4931

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1. Executive Summary

This project addresses the needs of process heater operators for increased energy efficiency, reduced emissions and improved operation and control. Process heaters in the U.S. consume about 3 quadrillion Btu annually. Conventional heaters operate at an efficiency of about 83 percent, and as a consequence energy losses constitute a large and excessive portion of total production costs. In parallel with the stress on energy efficiency, regions with high concentrations of process heaters have been brought under stringent emission control regulations. Attempts to reduce emissions using conventional combustion equipment resulted in degraded performance and product quality, and frequent capacity derate to avoid heater damage. To address these issues, the team of TIAX LLC, ExxonMobil Research and Engineering, and Callidus Technologies started in March, 2000 to develop designs and components for an integrated process heater technology that maximizes system performance (in terms of efficiency, emissions, flexibility, reliability and safety) while minimizing operational and maintenance costs. The performance target goals of the integrated technology were burner emissions of < 10 ppm NO_x (at 3 percent O_2) and < 5 ppm CO, and an overall heater efficiency of 95 percent. The intended benefits of this program are overall nationwide energy savings in the process industries, affordable emissions compliance for heater operators, and improved performance of fired heater equipment.

To develop individual components of the integrated system, the project initially pursued five parallel tasks: (1) ultra low emissions burner development; (2) Advanced fired heater design (3) Flame ionization system and air fuel ratio sensor (4) predictive emissions monitoring system (PEMS); and (5) Integrated Control System. Additionally, there was provision for a programmatic level effort on technology planning and analysis that supported each the individual tasks. After initial developments in the five primary tasks, the project scope was reduced down to three tasks: Ultra-low emission burners, Advanced fired heater design, and on-line non-intrusive, tube metal temperature monitoring. The task on flame ionization sensing generated data from subscale burner testing at TIAX that identified specific flame characteristics that showed control via flame sensing was technically feasible. Industry advisory groups, however, concluded that the cost and operational complexity of such a system would probably not be acceptable to heater operators. Also, the Callidus ultra low NO_x burner developed in parallel and now used industry wide did not require control based on flame ionization sensors. Similarly, the initial efforts in predictive emissions monitoring showed that correlations between process conditions and emissions could be developed, but such an approach to heater operation and emissions control was not of interest to industry due to incompatibility with regulations, compliance risk, and complexity. Continuous emissions monitoring was judged as more reliable and is currently used industry wide. For these reasons, the flame ionization system and PEMS tasks were truncated in 2002.

The low NO_x burner development started with the Callidus LE burner that achieved 30 to 40 ppm NO_x through fuel staging and induced flue gas recirculation. To identify and screen NO_x reduction concepts, TIAX adapted results from a versatile 2 MMBtu/hr burner fired in its test furnace, varying burner configuration and flame stabilization concepts. These results showed that emissions in the 5 ppm range were achievable with a single sub-scale burner. Callidus constructed a 9 MMBtu/hr full-scale burner for further optimization and design development in their test furnace. To support the burner scale-up, TIAX and Callidus conducted computational fluid dynamic (CFD) modeling to evaluate alternate burner design parameters. The Callidus full-scale burner tests showed emissions in the range of 6 to 10 ppm for a wide range of simulated refinery fuel gas compositions, with the higher emissions occurring at higher firing rates.

The third step in development was a field demonstration of 14 prototype burners in a 125 MMBtu/hr atmospheric pipestill furnace at an ExxonMobil refinery. The retrofit was done in parallel with CFD modeling to predict and correlate flame and emissions characteristics. The CFD modeling showed that burner swirl was important in achieving acceptable flame shaping. Operation with all burners set to swirl in the same direction gave erratic flame shapes with some wall impingement. Operation with adjacent burners set to swirl in counter rotating directions gave an overall stable flame envelope for the 14 burners, and no predicted impingement or high heat flux conditions. Initial tests showed emissions in the range of 15 to 17 ppm, but the burner exhibited pulsations when the methane content of the fuel was high. Accordingly, a series of flame stabilizers and gas tips were tested and a final stable design was selected for commercialization. This design yielded emissions of 22 ppm in the ExxonMobil demonstration heater. Callidus has commercialized this burner under the Callidus Ultra Blue (CUB) model designation. Through June 2006, 1550 of the original design CUB burners have been sold. In the interim, Callidus also developed 6 subsequent burner designs evolved from the original CUB development. Collectively, 2500 of these later generation burners have been sold by Callidus. Emissions with the CUB burners in the field have been in the range of 5 ppm – 30ppm depending on heater design and service, fuel composition, and firing rate.

The advanced fired heater design was developed through parametric design evaluations of a variety of furnace and combustion air preheater configurations and technologies for enhancing convective and radiative heat transfer. A prototype application for these evaluations was a 100 MMBtu/hr vertical tube box heater. Criteria used in the design evaluations were constructability issues, cost, potential for low emissions, and modular adaptability to a variety of alternate process applications. The design evolution relied heavily on computational fluid dynamic predictions of a parametric array of radiation and convection section tube design configurations, and burner placement alternatives. A significant effort was made in evaluating enhancements for radiative and convective heat transfer. In the end, the enhancement alternatives introduced some constructability complexity and adverse effects on flame shape for the advanced heater design while not contributing decisively to performance. Enhancements may however benefit performance with conventional heater designs.

The final advanced fired heater design has the following features:

- Modular radiant cells, each with one and two-side fired vertical tubes
- Vertical tube convection banks enclosed within the radiant cells.

- Integrated, commercial modular plate combustion air preheaters
- Forced draft fans (pressurized heater)
- Short stack

The predicted performance for the integrated advanced heater is summarized in Table 1-1 with comparisons to heaters with comparable product output.

Table 1-1. Estimated Performance of AFH

Heater Type	Efficiency % (LHV)	Heat Fired MMBtu/hr (LHV/HHV)	Total Erected Cost M$	NO_x Emissions lb/MMBtu (HHV) and tons/yr	CO_2 Emissions lb/MMBtu (HHV) and ktons/yr
Conventional fired heater	83	120/132	5.2	0.020/11.5	126/71.5
Conventional fired heater with air preheat	93	108/118	8.8	0.025/13	126/64.0
Advanced fired heater with air preheat	95	105/115	8.1	0.025/12.6	126/62.3

Capital cost savings = $700K (estimate performed July 2003), fuel cost savings based on $3.5/MMBtu = $92K/yr

Net present value of savings (15 yrs, 12% IR) > $1.0M per application

The efficiency target for the project of 95 percent is projected to be met by the AFH design. The NO_x emissions for the AFH correlate to about 12 ppm for refinery gas, slightly higher than the 10 ppm target. The total erected cost of the AFH is about 8 percent less than a conventional heater with combustion air preheat, and 56 percent higher than a conventional heater without combustion air preheat. ExxonMobil is currently taking required steps for licensing the AFH design.

The tube metal temperature monitoring development was done through a larger ongoing effort ExxonMobil was conducting with Honeywell to demonstrate application of low cost infrared camera technology to fired heaters. A feasibility study was completed in May 2003 showing the advanced movable thermal based IR detectors offered a low cost approach to tube metal temperature (TMT) monitoring in the heater. Subsequent development and testing was done on an ExxonMobil fired heater. The initial effort was on developing a reliable method for inserting and positioning the IR sensor in the furnace and transmitting data. This effort was successful. Subsequent testing showed that thermal gradients inside the sensor enclosure caused image banding that was not correctable. The overall concept and positioning hardware shows promise for on-line TMT monitoring, but a different imaging technique is needed for this application.

2. Project Summary

2.1 Objectives

This project focused on improved efficiency and reduced emissions for process heaters used in the petroleum refinery and chemical process industries. This equipment category was selected because it is a major energy consumption sector which has not extensively implemented energy efficiency technology, and it has recently been brought under very stringent environmental regulations. Table 2-1 shows energy consumption in the refinery and chemical process industries. Together, these two sectors account for 2,000 TBtu/yr, which is approximately 40 percent of U.S. energy consumption for total process heat. Natural gas accounts for approximately 30 percent of fuel use in refinery process heaters and 65 percent in chemical process heaters.

Table 2-1. Process Heater Energy Consumption

Refinery Processes

	Total Fuel Consumed TBtu//yr	Natural Gas Consumed TBtu/yr
Atmospheric distillation	351	104
Vacuum distillation	102	30
Catalytic cracking	156	46
Hydrocracking	50	15
Steam reforming	188	55
Hydrotreating	206	61
Catalytic reforming	208	61
Thermal cracking	9	3
Delayed coking	103	30
Total	1373	405

Chemicals Processes

	Total Fuel Consumed TBtu/yr	Natural Gas Consumed TBtu/yr	Natural Gas Consumed TBtu/yr	Fraction Natural Gas %
Ethylene	342	110	32
Ammonia	189	189	100
Carbon Black	40	38	95
Methanol	28	28	100
P-Xylene	25	23	90
Vinyl Chloride	17	17	100
Urea	16	16	100
Styrene	17	13	78
Benzene	9	8	90
Total	683	442	65

Despite the high energy consumption, this sector has not significantly benefited from advances in energy efficiency technology. Process heater designs have not appreciably advanced over the past 20 years. Over half of installed heaters are not equipped with combustion air preheaters for energy recovery from stack gases, resulting in a loss of 7 to 10 percent of energy consumption. This is primarily because the energy recovery technology was not judged cost effective at the time of construction for small and moderate duty process heaters.

In addition to the opportunity for energy savings, there is also a current need for environmental control, especially NO_x reduction. Refineries in Texas and California have been brought under stringent NO_x control regulations. The Houston-Galveston area, in particular, which accounts for 29 percent of US refining capacity, requires 87.5 percent NO_x reduction relative to emissions in 2000, and the trend is toward increasing stringency.

With increasing energy costs, and more challenging environmental regulations, energy conservation and environmental compliance are key business objectives for the oil industry. The objective of this project was to develop designs and components for an integrated process heater technology that maximizes energy efficiency, operational flexibility and reliability, while reducing emissions to compliance levels. The focus of the integrated system is on new heater construction. However, the emission reduction technology, and some of the design components in the advanced design, are also applicable to retrofit of existing equipment. The performance targets for the project were:

- NO_x emissions less than 10 ppm @ 3% O_2
- CO emissions less than 5 ppm @ 3% O_2
- Heater thermal efficiency: 95 percent based on lower heating value
- 15 percent discounted cash flow return on investment for incremental capital costs

2.2 Project Scope

The scope of work specified to address the project objectives consisted of the following five tasks

Ultra Low Emission Burner Development:

- Develop concepts for adapting conventional Callidus/TIAX low NO_x burner designs into an advanced ultra low NO_x burner capable of single digit NO_x emissions
- Conduct sub-scale parametric testing and adapt prior work of TIAX, Callidus, and GRI on fuel staging, premix, and internal flue gas recirculation concepts
- Perform CFD modeling design studies of single and multiple burners to support design, scale up and test data interpretation, and windbox design
- Scale up burner and test in a single burner test furnace
- Develop prototype ultra low NO_x burner design(s)
- Install and perform extended field demonstration in an operational refinery heater
- Modify burner as needed to optimize performance

- Commercialize burner and evolve design enhancements based on field results
- Adapt burner to the advanced fired heater design

Advanced Fired Heater Design:

- Evaluate overall concepts for heater layout in terms of efficiency potential, constructability, emissions, modularity, and cost
- Conduct CFD studies of alternate design configurations to support conceptual designs and evaluate heater performance
- Evaluate advanced concepts for heater components
  - Enhanced heat transfer in radiant and convection sections
  - Compact combustion air preheater designs
  - Convection section design
  - Modular packaging for flexibility
- Conduct detailed design and cost estimation
- Evaluate feasibility of field testing specific components of heater system
- Conduct parametric CFD studies of component alternatives to optimize the design
- Evaluate constructability issues and estimate costs
- Evaluate commercialization alternatives

Flame Ionization System and Air Fuel Ratio Sensor:

- Evaluate alternatives for FIS/AFR monitoring in refinery heater burners
- Conduct proof of concept testing in lab scale furnace
- Evaluate technical feasibility and cost for inclusion in integrated control system

Predictive Emissions Monitoring System:

- Evaluate commercial PEMS systems and variants as an alternative to continuous emissions monitoring, and as an on-line control of the burner for optimized low emissions
- Quantify technical effectiveness, reliability, flexibility, cost, and compatibility with emerging regulations
- Conduct market assessment through discussions with user groups
- Decide if PEMS has an effective role in the integrated design

Integrated Control System:

- Evaluate on-line tube metal temperature sensing technology for burner control to optimize heater performance and product quality
- Identify Candidates for development as part of integrated control system
- Conduct parametric developmental testing

- Conduct field testing in an operational refinery heater

In addition to these five technical tasks, DOE provided for a general support effort in Technology Planning and Analysis. The objective was to evaluate emerging and advanced technologies for energy efficiency and emission reduction in the process industries. Specifically, technologies were evaluated on the basis of energy or emission reduction potential, cost, developmental status, and total industry-wide energy savings potential. This task was intended to support both the efforts of the this project as well as other DOE programs.

2.3 Project Team

The team members and roles on the project are summarized below:

TIAX LLC

- Prime contractor and project manager for DOE interactions
- CFD modeling of burner and heater performance
- Combustion pilot scale testing and field testing diagnostics support
- Design and systems integration support
- Technology transfer

ExxonMobil Research and Engineering Company

- Process heater design and cost estimation
- Commercial system integration and testing
  - Low NO_x burners
  - Advanced heater components
- Heater diagnostics and control evaluation, testing and design
- Technical readiness assessment of AFH design
- Field demonstration planning and execution
- Process heater commercialization

Callidus Technologies

- Low NO_x burner development
- CFD design support
- Commercial scale burner testing
- Field support
- Ultra Low NO_x burner commercialization

In addition to this team, the firm of Norton Engineering supported the advanced heater design effort as a subcontractor to EMRE. Norton contributed to the heater and control system design, constructability evaluations and cost estimation.

2.4 Task Approach

The sequence of technical activities and the interactions among the five tasks specified in Section 2.2 are illustrated in Figure 2-1. The key activities, accomplishments and task interactions are summarized in the following sections. Technical results for the burner development, heater design, and tube metal temperature monitoring are discussed in Section 3.

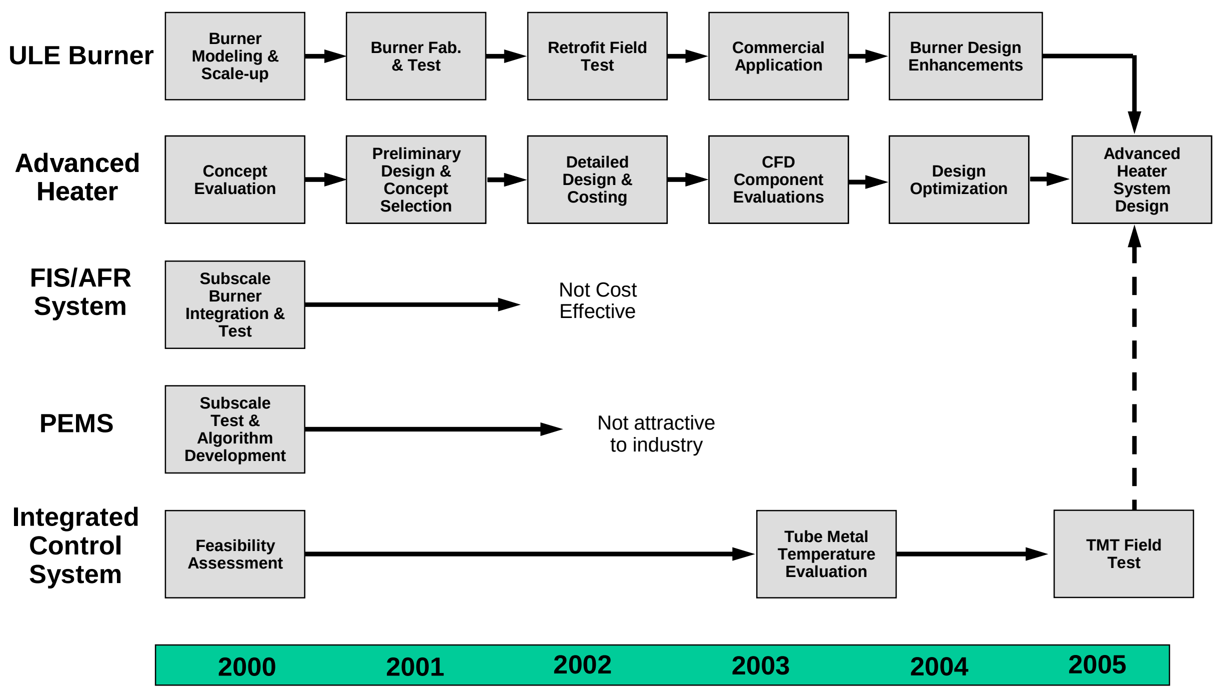


Figure 2-1. Project Approach

2.4.1 Ultra Low Emission Burner Development

Work was initiated on this task in January 2000, prior to contract start date of March, 2000, in order to meet ExxonMobil schedules for a refinery demonstration test in 2001. Two burner concepts were evaluated and developed in parallel: a fuel staging, internal flue gas recirculation burner, similar to conventional Callidus burners, and a lean pre-mix burner derived from work by Callidus and TIAX for Gas Research Institute prior to this project.

Initially, a specification for the low NO_x burner was developed jointly by TIAX, Callidus and ExxonMobil. A computational fluid dynamics model was introduced to facilitate design support and test data interpretation. To build and validate the model, initially the CFD runs were made for the sub-scale 2 MMBtu/hr prototype burner previously tested under a prior GRI contract. The CFD model was validated against the detailed in-flame species, temperature and velocity data generated at the Sandia Livermore Burner Engineering Research Laboratory (BERL). Excellent agreement was obtained, providing good confidence in the utility of the CFD approach for scale-up design.

The benchmarked computational fluid dynamics modeling tools were extended to the full-scale burner, and used to screen numerous design options to arrive at an initial 9 MMBtu/hr, experimental prototype burner design. A detailed mechanical design was developed for the experimental prototype burner and engineering drawings were generated. The burner prototype was fabricated, including several different options for the premix flame holder.

The first round of development testing was conducted at Callidus' test facility with support from TIAX and ExxonMobil. These initial tests showed NO_x emissions in the range of 6 - 10 ppmv (corrected to 3% O_2, dry basis) were achieved while firing with simulated refinery gas (30%H_2 / 45%C_1 / 25%C_3), at 6 MMBtu/hr and 3.2% O_2. Some burner instability was evident at certain operating conditions.

In parallel with the development of the non-premix burner, development proceeded on the advanced premix burner. The goal of the scale-up effort was to adapt the patented combustion process that served as the guiding principle for earlier sub-scale burner development. In this configuration, the combustion of a *lean*, primary mixture of fuel, air and internally recirculated flue gas was stabilized within the burner quarl by both mechanical and aerodynamic means, with the assistance of an adjacent premixed flame. When optimally configured for low NO_x operation the primary fuel, air and flue gas mixture was sufficiently inert that stable combustion could not be achieved without use of the premixed flame feature. Additional staged fuel is injected and combusted down-stream from the "quarl flame" in a way that delays combustion, entrains additional flue gas into the resulting flame volume, and allows the overall burner equivalence ratio to be near stoichiometric. Stable, natural draft, operation of this burner configuration was demonstrated from cold box start-up to full fire (9 MMBtu/hr) conditions encompassing the full range of excess air levels encountered during normal operation. NO_x emissions of about 8ppmv (corrected to 3% O_2, dry basis) were achieved when firing with simulated refinery gas at 9 MBtu/hr. For these tests the premixed flame holder was not optimally configured for lean operation.

Additional efforts were made to improve and broaden the premix burner stability. Alternative design options were evaluated for the premix flame holder. These included: 1) a perforated plate design that was flashback resistance with high hydrogen fuel contents during the development of the original 2 MMBtu/hr burner prototype, and is durable in the event of flashback, and 2) the low open area version of the "wiggle strip" style flame holder which prevents flashback for pure hydrogen fuel and air mixtures.

Due to remaining stability problems with the premix burner, and the necessity to begin fabrication of the full-scale burners to meet field schedules, the decision was made in September, 2000 to utilize the non-premixed version of the ultra-low-emissions burner for the initial tests installed in the ExxonMobil demonstration heater. This non-premix burner configuration had exhibited excellent stability while achieving NO_x emissions in the range of 7-11 ppm at typical heater firebox temperatures and ambient air. Although the premixed version showed potential to achieve lower NO_x emissions, its development lagged the non-premix burner, and it was not a reliable choice at that time. Development work continued on the premixed configuration during the field demonstration of the fuel staging burner.

In September, 2000, qualification tests were made at Callidus for the demonstration burner model. The test matrix included the following conditions: excess air varying over a three-to-one turndown range, low excess air, overfiring with 90 percent of the theoretical air requirement, and capacity turndown with 100 percent air flow. At key points, performance was monitored with three fuel blends (low, medium and high hydrogen content). The burner readily passed all of these tests and the ExxonMobil team qualified it for use in the selected refinery demonstration heater. Representative NO_x emissions measured during these tests are provided in Figure 2-2.

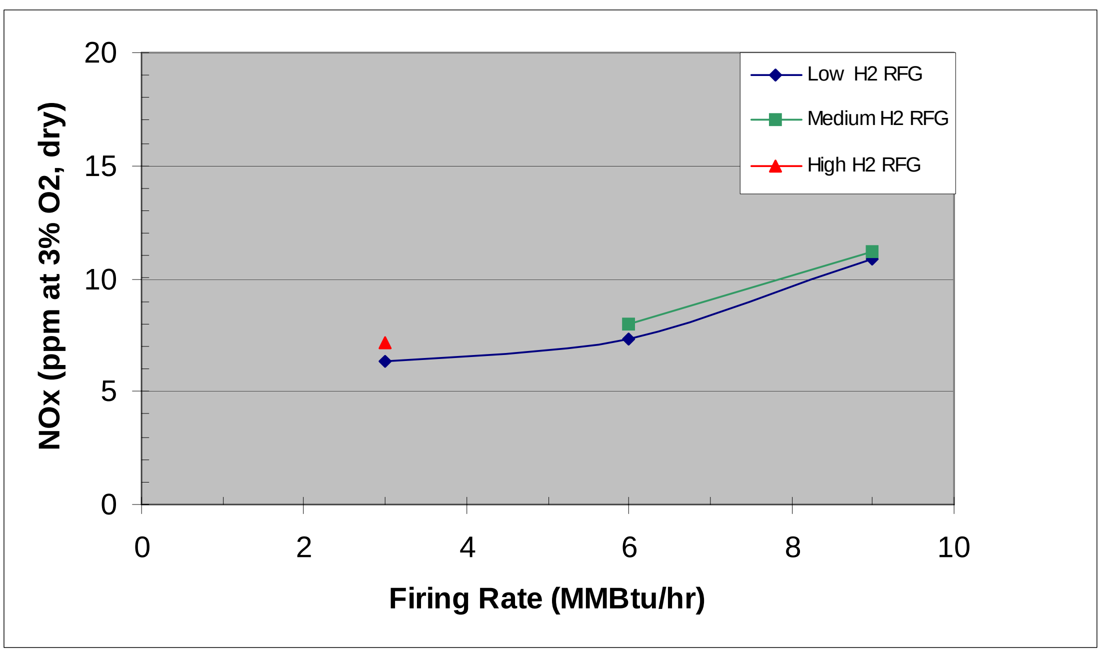


Figure 2-2. NO_x Emissions from 9 MMBtu/hr Burner

In October, 2000, baseline performance of the field demonstration fired heater was measured by the project team for the following parameters during normal heater operations: fuel composition, firing rate, stack oxygen concentration and temperature, emissions of NO_x and CO, bridgewall temperature, heat flux profile, tube metal temperatures, and process side parameters (flow, pressure, temperature).

The fourteen 9 MMBtu/hr first-generation ultra-low emissions process heater burners for the ExxonMobil heater retrofit demonstration were manufactured at Callidus in November, 2000. Burner test results of the first-generation burner were provided to 40 members of a Burner User Group at the Callidus test facility.

In parallel with the demonstration of the burner at the ExxonMobil refinery, Callidus commercialized the burner and three other refinery applications were initiated. These applications subsequently showed comparable performance to the prototype demonstration and qualified the burner for industry-wide commercial sales.

Prior to the field demonstration burner retrofit, a 3-D computational fluid dynamics (CFD) model was developed to give an early indication of how the burners would perform in the demonstration furnace. The CFD model provided flame shapes for each burner as well as overall flue gas flow patterns and temperatures in the heater box. These predictions were used to identify flame configuration problems and to suggest and qualify fixes for any field performance problems that could occur. Results showed potential for unacceptable heater operation due to flame leaning in some of the 14 demonstration furnace burners and possible antagonistic interaction of adjacent flames. The first model of the heater (14 demonstration burners positioned in a single row in the center of the heater firebox) showed potential for poor end burner flame patterns. These flames tended to lean towards the tubes, which would potentially cause tube overheating problems. The model indicated that the swirling motion generated by the burners was responsible for the flow pattern in the firebox leading to end burner flame lean. Additional CFD models were run wherein (1) the swirl direction in every other burner was reversed and (2) the swirl direction in only the end burners in the row were reversed. Both of the changes showed marked reduction in the flame lean tendency. Since opposite swirl in every other burner was judged to yield slightly better results, the burners were installed with this configuration.

The credibility of the modeling techniques and results mentioned above was verified by modeling the performance of other heaters using the new Callidus burner for which data were starting to become available. One heater that was modeled had a vertical cylindrical configuration with eight burners arranged in a circle. The CFD model predicted burner-to-burner flame interference and a fuel rich zone in the central core of the burner circle when all burners were in service. When some burners were turned off, the model predicted improved performance. These predictions correctly correlated with flame interaction behavior actually experienced in the field for this heater.

The ExxonMobil demonstration heater was retrofit in May, 2001 and subjected to emissions and performance testing. Initial data showed NO_x emissions as low as 15 ppm at 3 %O_2 (dry). As predicted by the CFD study, no leaning flame problems were observed. However undesirable combustion oscillations occurred when the percent methane in fuel gas unexpectedly exceeded the original design basis of 45 percent. It was subsequently determined that the fuel gas to this heater could at times contain up to 85 percent methane. Gas tip designs were modified and installed and these eliminated the combustion oscillations. However with the new tips, NO_x emissions increased to as high as 30 ppm at 3 %O_2 (dry) when operating at design heat release.

In parallel with the field demonstration, the team conducted a series of optimization tests on the commercial version, the CUB (Callidus Ultra Blue) low-NO_x burner in the Callidus test furnace in Tulsa. The objective of this work was to finalize the design of a new flame holder and gas tips which would provide improved performance relative to the currently installed burners in the demonstration heater.

The initial burners in the demonstration heater had a standard flame holder with modified gas tips. The modified gas tips were retrofitted in July 2001 to eliminate combustion oscillations experienced when large amounts of natural gas were injected into the fuel system. The modified tips were installed as an interim measure, recognizing they would increase NO_x and marginally reduce combustion air flow, until an optimum design could be developed.

During initial efforts to optimize burner performance, it was concluded that changes to the burner flame holder would be required to achieve the desired objectives. With flame holder changes, lower-NO_x-producing gas tips could be used with higher CH_4 content in the fuel gas. The new flame holder and gas tip design were finalized in November, 2001. A test burner with the new design was run through a series of tests at Callidus. These test results confirmed the design achieved the desired performance objectives with respect to NO_x emissions and flame stability. The following performance characteristics were observed:

- *Reduced NO_x:* At design heat release and excess O_2, NO_x measured in the test furnace was about 14 ppm (versus 20 ppm for the initial burner) when firing average hydrogen (55%) fuel gas and with the test furnace firebox temperature near 1800F. With other fuels, NO_x varied from 13 ppm with 100 % natural gas fuel to 15 ppm with high hydrogen fuel (80 % H_2, 13.5 % CH_4 and 6.6 % C_3H_8).

- *Increased combustion air flow:* At design heat release and O_2, the air side pressure drop was reduced from 0.4 to 0.33 inches W.C.

- *Stable combustion:* At the fuel basis of 60 percent maximum natural gas injected into typical refinery blend gas (equivalent to 71 percent CH_4 in the fuel gas mixture) excellent flame stability was achieved. The burner could easily be lit off in a cold firebox with air damper wide open when firing 100 percent natural gas and low hydrogen fuel. After light off the fuel pressure was adjusted to 1 to 2 psig to simulate start-up conditions. The burner performed acceptably under these conditions.

- *Low CO emissions:* CO emissions were essentially zero at all key operating points.

- *Eliminated intermittent burning under flame holder*

- *Acceptable flame shape:* Lower swirl levels in the primary combustion zone reduced burner-to-burner interactions and potentially reduced NO_x. The visible burner flame length at design heater release and O_2 was about 10 to 11 feet. CO measurements in the test furnace confirmed this based on a noticeable decline in CO content above this elevation. Total CO burnout (< 10ppm) occurred about 19 feet above the burner.

The new flame holder design featured the following modifications relative to the standard flame holder initially installed in the prototype demonstration burners:

- The slots in the “trough” of the flame holder were eliminated. This change eliminated the possibility of burning under the flame holder. The trough width was decreased to increase air side capacity.

- Vertical plates were added to the trough to reduce swirl and enhance stability.

- Vertical slots were cut into the full height of the center, cylindrical portion of the flame holder to enhance stability.

- The width of the "internal stabilizer ring" portion of the flame was reduced and a segmented ring, instead of a continuous ring, was used.

Based on these results, burners with modified flame holders and gas tips were installed in the demonstration heater. These changes yielded a noticeable improvement in flame shape, a modest reduction in NO_x emissions, and an improvement in flame stability. The heater has subsequently operated satisfactorily over four years with the modified burners.

The premixed burner configurations tested in parallel with the demonstration of the non-premixed burner had not yielded significant reductions in NO_x relative to the initial burner configuration (Callidus’ commercial CUB round flame model). Further development at that point focused on promising variations on the CUB configuration. One design that was evaluated employed a smaller burner diameter that produced a shorter flame. Both features are desirable with burner retrofit applications.

During and following the successful demonstration of the ultra low emission burner in the ExxonMobil field test, Callidus commercialized the CUB burner and sold a total of 1553 burners by June, 2006. Following the commercialization of the CUB round flame burner, Callidus continued development of variants on the burner either to broaden the addressable market, or to enhance performance. Six burner designs, evolved in part or totally from the original CUB development, have been developed since 2002. The burner designs, and the sales totals to June, 2006 are as follows:

- CUBF: Flat Flame — 256 burners sold
- CUBL: Small Diameter Burner — 849 burners sold
- CUBLF: Small Version of CUBL — 1307 burners sold
- CUBX: Lean Premix Burner — 16 burners sold
- CUBLX: Advanced Lean Premix Burner — 30 burners sold
- CUBR: Retrofit CUB Package — 31 burners sold

Cumulative sales of the six later generation designs are 2489 burners. Total sales of the original CUB and later descendant burners are 4042 burners, to June 2006. Sales of the original CUB design have ceased, but Callidus continues to sell the later generation burners.

2.4.2 Advanced Fired Heater Design

The design development for the advanced fired heater started in October, 2000 with an assessment of alternative technologies for the key components in the heater. The initial focus was on heat transfer enhancements. A preliminary comparative analysis of dimple tubes as a potential alternative for finned tubes in the convection section with respect to effective heat transfer surface, pressure drop and cost was performed and a literature survey of similar alternative sources of enhanced heat transfer surfaces was initiated. Other potential features reviewed were enhanced surface convection section tubes, emissivity enhancement of radiant tubes, and reradiating or reflecting surfaces for enhancement of radiant section heat transfer. Geometry variations evaluated included:

- reduced spacing between radiant tubes and refractory surface
- reduced spacing between the adjacent tubes
- single and multiple longitudinal fins on the back side of the radiant tubes
- concave refractory surface behind the tubes

In parallel with the study of heat transfer enhancements, overall configuration options were evaluated. Alternatives for optimum arrangement of the radiant section, convection section and combustion air preheater along with efficient use of space and material were identified and screened.

The single most important component in terms of efficiency improvement is the combustion air preheater. A cost break down of fabricated material and erection of air preheat systems was developed. A literature search was performed and several plate-type compact heat exchangers for air preheating were identified. In addition, alternative combustion air preheater designs (other than conventional regenerative and recuperative types) were surveyed and evaluated.

Parametric process engineering calculations of an optimal arrangement of the advanced heater configuration were made. Several design alternatives that can achieve the efficiency target level of 95% were evaluated from constructability and cost aspects. A standard cost evaluation template was developed for comparing the relative merits of competing technologies. Templates were developed for both retrofit of individual components to existing heaters, as well as application to the integrated advanced fired heater configuration.

A design review in August, 2001, highlighted the following key design features and issues:

- A vertical box (single or twin cell, depending on size) is the most compact design consisting of conventional fired heater components and air preheater technologies. Radiant section heat transfer enhancements were candidates to significantly improve heat flux uniformity and thereby reduce required tube surface area.
- Elimination of the induced draft fan can yield an improvement in system reliability, but requires designing and operating the heater at up to 10"w.c. pressure.
- Maintenance and inspection access to convection tubes must be incorporated into the design.
- Optimizing thermal efficiency by limiting system radiation loss to less than 1% and improving the radiant efficiency by coating refractory surfaces may be practical and economical. The largest potential for thermal efficiency improvements here are with retrofit of existing fired heaters.
- Compact cross-flow combustion air preheaters appear to be the most attractive configuration for the advanced fired heater application.
- Dimple tube technology for applications where bare-tube convection sections can be replaced by dimple tubes may yield about 2-3 % increase in thermal efficiency in units with large approach temperatures but less than 1% in those with close approach temperatures.

A CFD model of the advanced fired heater configuration was built. Analysis of a few selected configurations incorporating the above concepts was initiated in October, 2001. These results, as discussed in Section 3, showed CFD was a powerful tool for quickly and reliably screening candidates in design development.

Based on the conceptual designs and CFD results, a modular twin-cell, vertical-tube radiant box configuration having two-side-fired tubes in the center was selected in December 2001. The use of surface radiant heat transfer enhancements was deferred pending further CFD evaluation. With this configuration, flue gas exiting the radiant section passes through vertical tube convection sections (with extended convective surface heat transfer enhancements) and compact cross-flow combustion air preheaters.

The convection section design was further refined to include modularization. This approach enabled side-wall reinforcement through the additional structural steel used to form the modules. As a result the vertical refractory wall separating the radiant and convection sections could be fabricated without the need for additional internal support.

Preliminary specifications for the convection module outlets and hot and cold air ducting arrangements were developed for use in obtaining information from vendors/fabricators of compact heat exchangers. Vendors provided equipment arrangements, performance estimates and costs for combustion air preheaters for the advanced fired heater configuration. These data were used in screening alternate design configurations and in cost estimation for the integrated AFH.

CFD runs for the initial selected heater design indicated flame impingement in some sections, and high localized heat fluxes. To resolve these issues, a revised design for the modular, multi-cell vertical-tube radiant box configuration was completed. The radiant box height was decreased and the floor area increased to influence flow patterns to yield straighter flames and improved heat flux distribution. CFD model predictions showed the modifications were successful in yielding straight flames and reduced high heat flux. The CFD model results for the revised configuration indicated that all flames were relatively straight with no impingement on tubes, that heat flux levels were within acceptable ranges and that low flue gas temperature at the heater floor were conducive to low NO_x levels. Two modular, compact air preheater configurations were investigated. Each could be readily integrated with the advanced process heater. Manufacturing methods and costs were assessed for both retrofit configurations and for the advanced heater design. Two manufacturers were identified for production of the combustion air preheaters and cost estimates were obtained. The selected combustion air preheating system was based on commercially proven technology that was integrated in a unique way with the advanced heater.

An analysis of direct costs for construction of the advanced heater was completed. The results showed that the advanced configuration was approximately the same cost as a conventional process heater equipped with a combustion air preheater.

Further CFD analyses with the use of extended heat transfer surfaces in the radiative section indicated that the surfaces could induce flame leaning and possibly higher heat fluxes on the center tubes. The incremental increase in efficiency was small, and did not warrant the increased cost and manufacturing difficulty. Accordingly, it was decided in July, 2005, not to include heat transfer enhancements in the radiant section of the advanced fired heater. The use of enhancements may, however, produce significant performance improvement potential for existing fired heater designs, with heater configurations different from the AFH. The possibility of evaluating the effectiveness of enhancements for retrofit to an existing heater was considered but not pursued due to funding limitations.

After selection of the final design, a design manual was prepared for the individual components and for the integrated AFH. EMRE subjected the design to a Technical Readiness Assessment Considerations (TRAC) process. The TRAC process was nearing completion as of June, 2006, and at that point, the design was approaching technical readiness for deployment. EMRE is taking required steps for licensing the AFH technology.

2.4.3 Flame Ionization System (FIS) and Air Fuel Ratio Sensor

Initial evaluations focused on evaluating flame spectroscopic characteristics to determine if a reliable and repeatable flame signature could be used for burner control.

In March 2000, the TIAX combustion test furnace and data acquisition systems were used for testing of a pilot-scale 2 MMBtu/hr burner with provisions for the acquisition of the FIS signal. The burner used for the tests was the low NO_x pilot scale burner used for the earlier development of the Task 1 burner development.

The 2 MMBtu/hr burner prototype was successfully fired in the TIAX test facility and a strong FIS signal was obtained from the premixed flame. After a series of tests, the measured FIS signal was correlated against the burner's premixed and overall equivalence ratios over a range of firing rates and excess air levels.

Results indicate a two-electrode configuration produces a strong signal that varies with key burner operating parameters such as firing rate, overall excess air level, and premixed equivalence ratio. These data were evaluated to determine if the trends and sensitivities are appropriate for monitoring and control purposes. Results indicated that the patented log-slope method of determining air-fuel ratio can be applied to the premix zone of the burner. In addition, results showed that the variability in the FIS current signal can be used to provide an indication of approach to the lean stability limit. After these tests, it was concluded that use of the FIS signals would not be a cost effective addition to the integrated control system, and no further work was done. Also, the burner development effort in Task 1 was evolving toward using a non-premix burner technology, which did not require FIS-based control for stable operation.

2.4.4 Predictive Emissions Monitoring System (PEMS)

At the outset of the project, it was determined that PEMS would not be an effective component of an integrated control system for refinery heaters. The technology exists for predictive monitoring, but discussions with burner user groups indicated that PEMS were perceived as less accurate and less reliable than continuous emissions monitoring for complying with stringent emission regulations. Additionally, the Burner Users Group (BUG) thought that evolving regulations in districts with high heater populations would not accept PEMS in the future. No further work was conducted on this task.

2.4.5 Integrated Control System

Initial evaluations of tube metal temperature sensing indicated that improved reliability and on-line performance diagnostics could be technically feasible. Discussions with burner user groups and refinery heater operators indicated that active combustion controls based on TMT measurements would be more complex than other heater system components, and would not be cost effective. It was decided to pursue TMT sensing that would be real-time data provided to operators, but would not be an active part of the combustion controls. In this way, the operator could run the heater at an efficient, low emission condition and would receive warning if incipient coking or tube over temperature was occurring. Most of the benefits of on-line TMT would be accrued without the additional cost.

Initial evaluations in this task showed that there were low cost infrared cameras that showed potential for more accurate and comprehensive TMT measurements than conventional thermocouples or other intrusive techniques. In 2003, a parallel related program jointly conducted by ExxonMobil and Honeywell was identified. The effort in this task was merged with the ongoing ExxonMobil and Honeywell effort. Initial proof of concept tests were done in early 2003. On that basis, the second step of testing the system in an operating heater at an ExxonMobil refinery was started. Initially the design and operation of the camera positioning equipment was tested. After a series of design refinements, this effort proved successful. Subsequent testing of the camera in the furnace showed that a series of baffles were necessary to avoid reflections of images into the camera that interfered with the intended signal. Further tests led to the conclusion that the infrared camera used in the tests was not adaptable to the high thermal gradients of an in-furnace enclosure. The camera exhibited image banding and no acceptable means was found to compensate or correct for the banding. It was concluded that the overall concept of in-furnace IR thermometry was valid for heater applications, but a different camera design was needed. No further work was scheduled following conclusion of the field test in June 2005.

2.4.6 Planning and Analysis

In addition to the directed technical tasks discussed above, DOE added provision to the contract for short, focused technology support tasks dealing with process combustion technology planning and analysis. Two tasks were conducted during the project to evaluate emerging and potential technologies for improving efficiency, process performance, or emissions in process heaters and furnaces. The tasks identified five categories of technologies that were considered to offer the best combination of high to moderate improvement in performance, low to moderate development risk and diverse applicability across the range of process heat applications. The technologies cited were:

- Advanced heat recovery: development of enhanced heat exchangers, and development of add-on waste energy cycles

- Burner control: flame shape, capacity and fuel/air distribution determined by feedback on flame spectroscopy, process thermal performance, and emissions
- Process heater sensors and controls: improved diagnostics and control of product quality
- Universal heating system: modular combustor designs adaptable to a wide range of integrated, optimized furnace designs
- Oxy fuel systems: adapt hardware developed in other oxy fuel developments to process heating applications

The total annual energy consumption in industries addressable by these technologies is 14.4 quads. The study estimated that implementation of the above technologies would realize a 12 percent reduction in process heat fuel consumption, approximately 1.7 quad annually. Cumulative estimated energy savings through 2030 was estimated at 17 quads.

2.5 Technology Transfer

During the project, the following technology transfer activities resulted.

2.5.1 Publications

The following papers were prepared and presented on the results of this project:

High Efficiency, Ultra-Low Emission Process Heater" Benson, C., Loftus, P., and Juedes, D. presented at the Petroleum Environmental Research Forum, March 8, 2000

"High Efficiency, Ultra-Low Emission Process Heater" , Benson, C., Loftus, P., Pellizzari, R., Juedes, D., Chhotray, S., Bishop, D., and Martin, R. presented at the 2001 Joint International Combustion Symposium, September 8 – 12, 2001, Kauai, Hawaii

"High Efficiency, Ultra-Low Emission Process Heater," Benson, C, Loftus, P. and Juedes, D. presented at the International Energy Agency Meeting, Trondheim, Norway, June 23 – 26, 2002

"Numerical Assessment of Extended Surfaces in the Radiant Section of a Vertical Cylindrical Heater" , Boral, A., Sinha, J., and Mason, H. Presented at the American Flame Research Committee Fall 2005 Symposium, November 8, 2005, Atlanta, GA.

2.5.2 Patents

The following patent was issued during the project incorporating technology developed in the project:

Method and Apparatus for Heating a Furnace, Benson, C.E., Pellizzari, R.O., Loftus, P.J., Juedes, D.L., Martin, R. R., Wade, E.R,, and Rodden, P. M., U. S. Patent 6,672,858, issued January 6, 2004.

In addition, TIAX and EMRE are in the process of applying for a patent on the integrated advanced fired heater design.

3. Technical Results

This section provides details on the technical analyses and results for the task approach described in Section 2.

3.1 Ultra Low Emission Burner

3.1.1 Burner Development

The target applications for the advanced burner are process heaters in the chemicals and refining industries, which together account for 2000 TBTU/yr. of energy consumption in the U.S. The benefit of the advanced burner technology is to provide operators an option for compliance with new regulations while preserving or improving heater performance.

United States regulations requiring lower NO_x emission levels have become increasingly difficult to meet technically and economically. In the Houston-Galveston area of Texas, the State Implementation Plan (SIP) calls for phased reduction of NO_x from point sources with final reductions potentially as high as 90% relative to the year 2000 baseline. There are stringent emission specifications that are used in conjunction with firng rates to set final NO_x allocations. For example, for gas fired heaters > 100 MMBtu/hr and gas fired boilers, NO_x emission specifications are 0.010 lb/MMBtu (~ 9 ppm). The emission specifications are slightly less stringent for smaller units. For example, for gas fired heaters between 40-100 MMBtu/hr, NO_x emission specifications are 0.015 lb/MMBtu (~13 ppm). With proposed cap-and-trade programs, there is also increased incentive for further NO_x reductions, if they can be achieved at relatively low cost.

At the start of this project, burner technology generally could not meet these requirements cost effectively. Existing ultra-low-NO_x burners had not attained NO_x emission levels of 0.015 lb/MMBtu or lower. Combustion modifications such as external flue gas recirculation (FGR) and steam injection could approach this level, but generally could not assure compliance and had severe operational limitations. External FGR usually requires high capital investment for fans and ducting and reduces thermal efficiency. Steam injection sacrifices plant efficiency. Post-combustion controls, such as selective catalytic reduction (SCR) and selective non-catalytic reduction (SNCR), cannot meet required levels at low cost. Accordingly, the design criteria set for the ultra low emission burner were:

- Lower NO_x emission levels compared to existing, available burners over a wide range of heater designs;
- Safe operation with minimal constraints on normal operating ranges
- Adaptable to either the advanced fired heater or to retrofit applications;
- Costs significantly less than alternatives, specifically SCR.

The advanced low-NO_x burner that was developed was based on Callidus' successful line of staged-fuel, internal flue gas recirculation low-NO_x burners (Model LE,

Figure 3-1). The staged-fuel approach separates the combustion process into two zones. In the first, all of the combustion air is supplied along with a fraction of the fuel and internally recirculated flue gas. The remaining fuel and additional internally recirculated flue gas are introduced downstream into the secondary zone. In the advanced low-NO_x burner, proprietary design elements have been incorporated to enhance internal recirculation and improve mixing while maintaining required combustion stability. As a result, flame temperatures are relatively uniform and well below the threshold (2800°F) where thermal NO_x formation becomes significant.

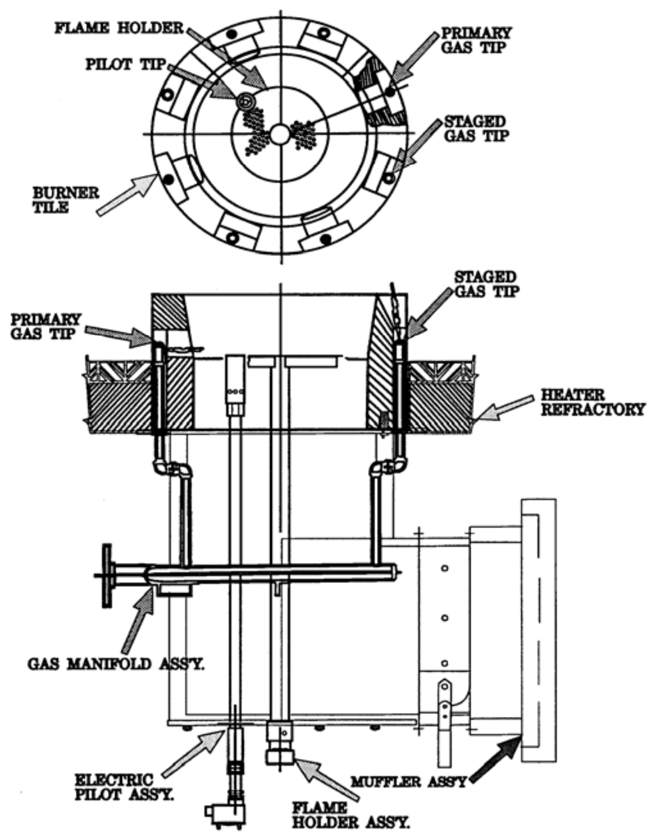


Figure 3-1. Callidus LE Burner

In the initial step of the DOE program, the burner was modified to reduce emissions and advanced to commercial readiness. The starting point was a sub-scale 2 MMBtu/hr prototype (Figure 3-2) that was developed previously at TIAX's industrial combustion test facility (Figure 3-3).

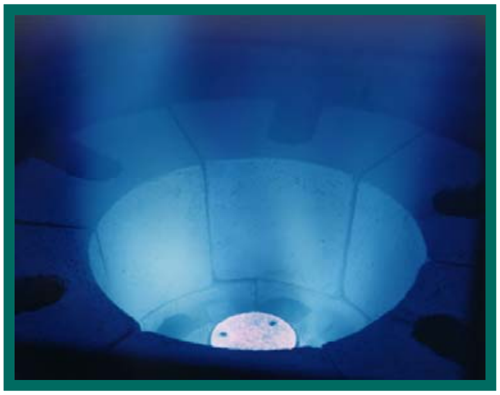


Figure 3-2. Prototype 2 MMBtu/hr burner

Figure 3-3. TIAX Combustion Laboratory

The 2 MMBtu/hr natural draft burner demonstrated very low levels of NO_x emission (Figure 3-4). It was tested using natural gas and simulated refinery fuel gas (RFG). The following was demonstrated:

- NO_x emissions from 4-6 ppm (corrected to 3% O_2, dry basis)
- Turndown ratio of 8:1 using natural gas
- Turndown ratio of 5:1 using RFG
- Turndown ratio of 3:1 with full airflow
- CO emissions less than 5 ppm
- PAH emissions 1-2 parts per trillion
- Pressure drop of 0.35-0.4 in. w.c.

At the outset of the DOE project, a modified, larger 9 MMBtu/hr version was built to demonstrate the technology in a single burner, water-cooled test furnace at Callidus (Figure 3-5). NO_x emission levels were 6-11 ppm when operating on a variety of RFG compositions with ambient temperature air and various heater flue gas exit temperatures up to 1800 °F (Figure 3-6). The design air-side pressure drop for the burner was 0.30 in. w.c. When tested as a forced draft burner, heat releases of up to 15 MMBtu/hr were

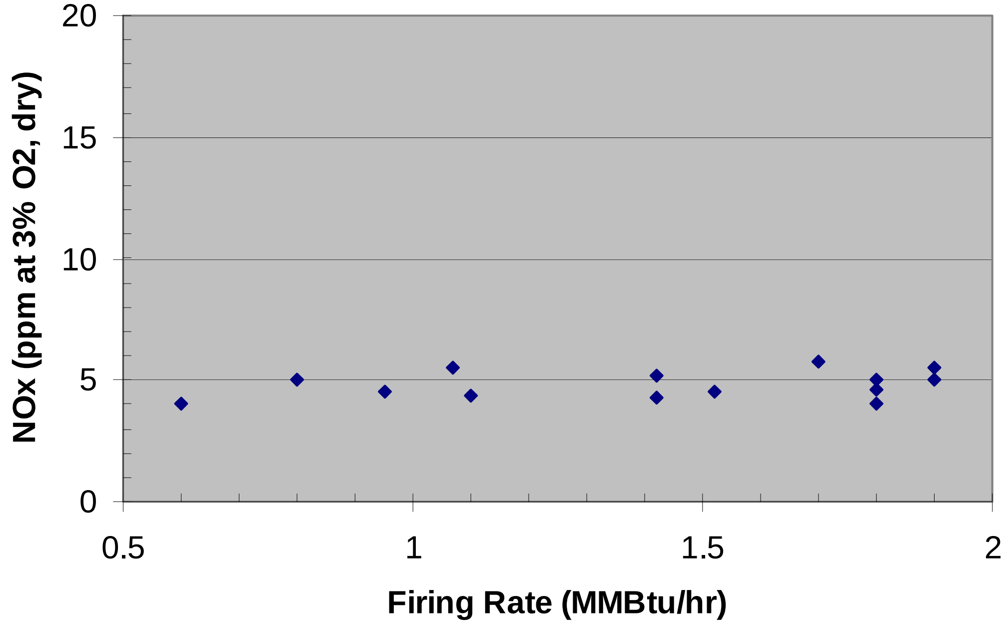


Figure 3-4. 2 MMBtu/hr Prototype NO_x Emissions

Figure 3-5. Callidus Test Furnace

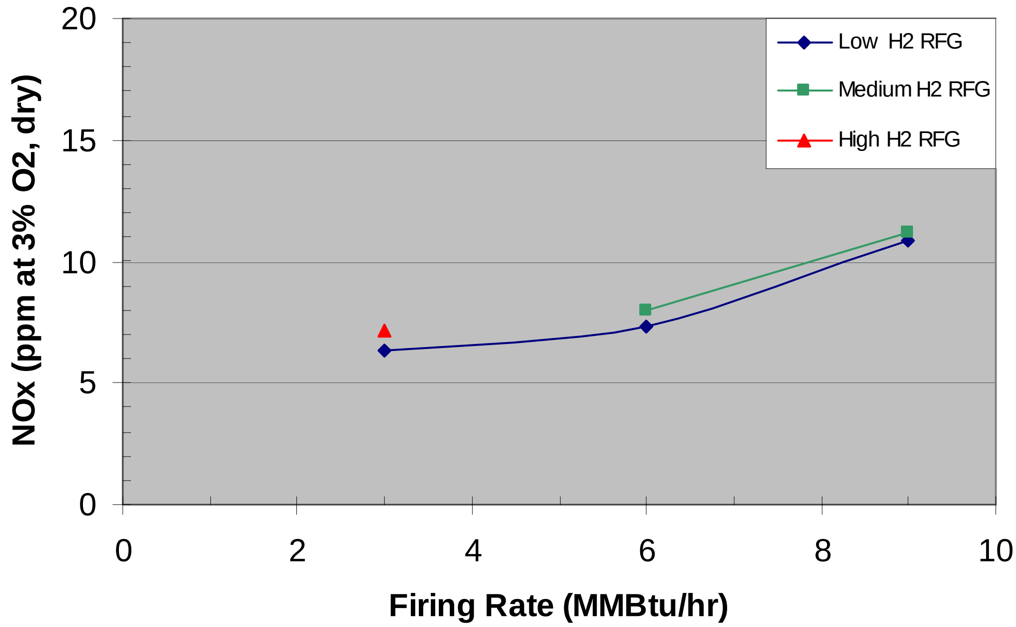


Figure 3-6. NO_x Emissions from 9 MMBtu/hr Burner

attained and NO_x emissions were approximately the same. Combustion air preheating was also tested. As expected, NO_x emission levels increased by about 40 % with 600°F combustion air.

Following successful design and demonstration of the single commercial scale burner, a field demonstration was conducted on an operational refinery heater at an ExxonMobil refinery. The demonstration heater had a horizontal tube cabin configuration and was used to preheat feed to an atmospheric pipestill. Consistent with many of ExxonMobil's heaters, this unit operated with natural draft and ambient temperature combustion air.

Prior to completing the burner retrofit design specification, a computational fluid dynamics model was utilized to verify that acceptable flow patterns and flame geometry would be produced. The model and predicted results are discussed in Section 3.1.2. The CFD results clearly show that the original design generates a large-scale swirling pattern in the furnace, leading to deflection of the end burner into the tube bank. Reversing the swirl direction of every other burner eliminates the large-scale swirl and yields relatively straight flames.

In May 2001, fourteen 9 MMBtu/hr advanced low-NO_x burners were installed in the pipestill furnace. Initial data showed NO_x emissions as low as 15 ppm at 3 % O_2 (dry). As predicted by the CFD study, no leaning flame problems were observed. However undesirable combustion oscillations occurred when the percent methane in fuel gas unexpectedly exceeded the original design basis of 45%. It was subsequently determined that the fuel gas to this heater could at times contain up to 85 % methane. Modified gas tips were quickly designed and installed and these eliminated the combustion oscillations. However with the new tips, NO_x emissions increased to as high as 30 ppm at 3 $\%O_2$ dry when operating at design heat release.

Burner Commercialization

In parallel with, and following the field demonstration, Callidus advanced the project's burner technology to the point of commercial sales under the trade name Callidus Ultra Blue (CUB). Figure 3-7 shows the commercial burner flue gas and fuel flow patterns. A cut away view of the burner is shown in Figure 3-8. About 1550 burners of the original CUB design have been sold to a variety of oil and chemicals companies. NO_x emissions from these units are in the range of 5 – 33 ppm. Section 2.4.1 provides details on commercialization of succeeding CUB burner design embellishments.

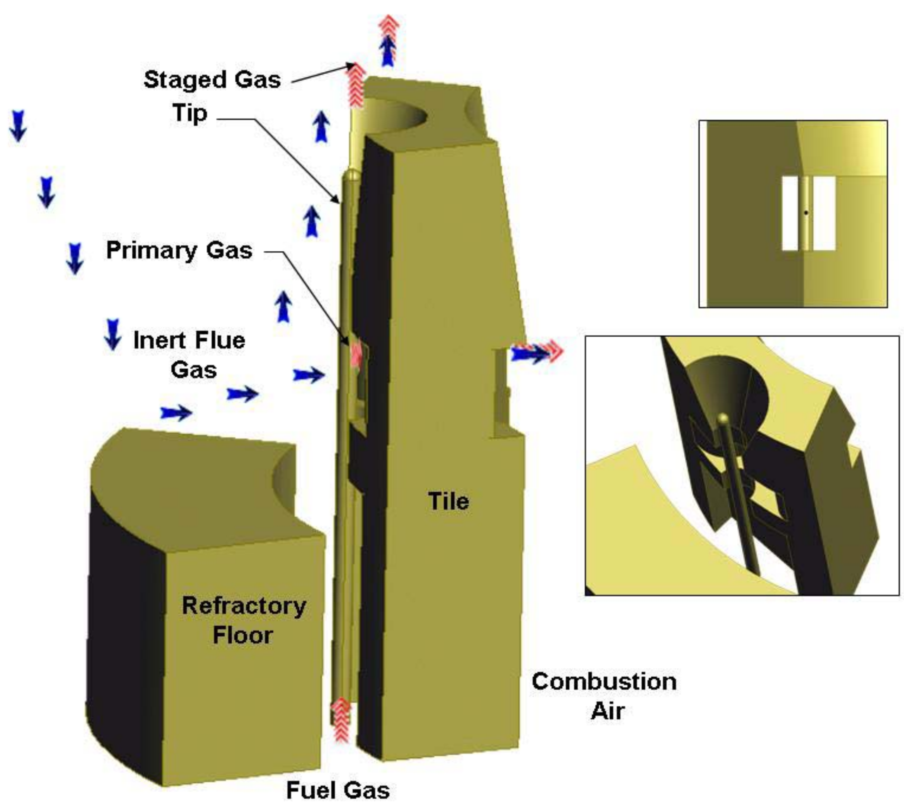


Figure 3-7. Flow Paths in Callidus Burner

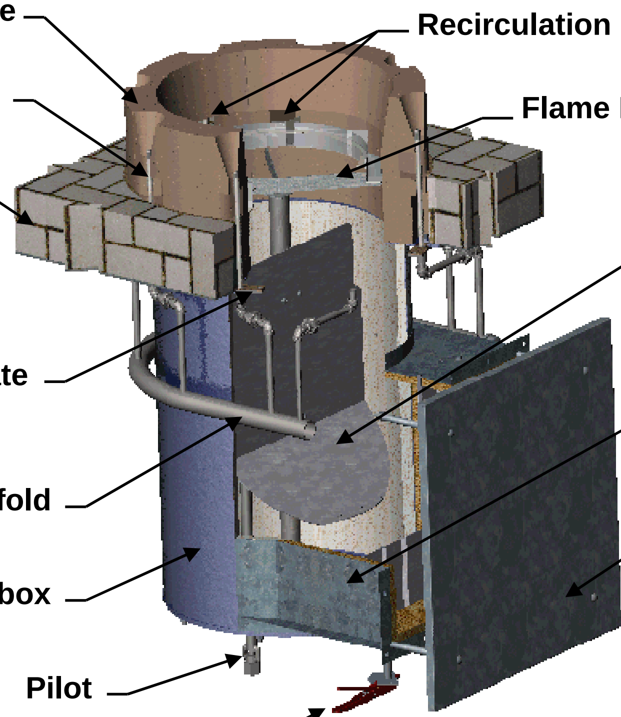


Figure 3-8. Callidus CUB Burner

3.1.2 Computational Fluid Dynamic Modeling

Computational fluid dynamic (CFD) modeling has found increasing use in the design and evaluation of combustion systems and emissions reduction technologies for fired heaters. TIAX's experience in the CFD modeling for industrial combustion systems was used extensively in the design and optimization of both the low NO_x burner and the advanced fired heater. The CFD analyses performed during this project enabled us to simulate various technologies for ultra-low-emission combustion, advanced heat recovery, and reduced energy losses in the virtual realm before finalizing the design. Following demonstration testing, the CFD models were used to interpret, correlate and extrapolate test data. This use of such three-dimensional, reacting flow simulations resulted in substantial savings in time and money during the entire span of the project. The CFD simulations performed during this program were performed using the latest version of FLUENT, a commercial general-purpose CFD code.

The flow field for both the burner and the advanced heater modeling was assumed to a steady-state, turbulent, reacting continuum field that can be described locally by general conservation equations. The governing equations for momentum, energy, turbulence, thermal radiation and chemical species were solved in an Eulerian framework. The Favre-averaged Navier-Stokes equations in Cartesian tensor form are given by,

$$\frac{\partial}{\partial x_i}\left(\rho u_i\right)=0$$

$$\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right)=-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}-\frac{2}{3}\delta_{ij}\frac{\partial u_m}{\partial x_m}\right)\right]+\frac{\partial}{\partial x_j}\left(-\rho\overline{u_i' u_j'}\right)$$

The turbulent heat transport was modeled using Reynolds analogy to turbulent momentum transfer.

$$\frac{\partial}{\partial x_i}\left[u_i\left(\rho E+p\right)\right]=\frac{\partial}{\partial x_j}\left[\left(k+\frac{c_p\mu_t}{\mathrm{Pr}_t}\right)\frac{\partial T}{\partial x_j}+u_i\left(\tau_{ij}\right)_{eff}\right]+S_h$$

The term S_h in the energy equation represents the heat of chemical reaction. The term with the deviatoric stress tensor, $(\tau_{ij})_{\mathrm{eff}}$, represents viscous heating.

$$\left(\tau_{ij}\right)_{eff}=\mu_{eff}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\frac{2}{3}\mu_{eff}\frac{\partial u_i}{\partial x_i}\delta_{ij}$$

The SIMPLE algorithm was used for pressure-velocity coupling. Turbulence was modeled using the two-equation k-epsilon turbulence model. The Boussinesq hypothesis was used to connect the Reynolds stresses in the momentum equation to the mean velocity gradients.

$$-\rho\overline{u_i' u_j'}=\mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\frac{2}{3}\left(\rho k+\mu_t\frac{\partial u_i}{\partial x_i}\right)\delta_{ij}$$

Transport equations for the turbulence kinetic energy (k) and the dissipation rate (ε) were modeled with the following rate equations,

$$\frac{\partial}{\partial x_i}\left(\rho k u_i\right)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\varepsilon$$

$$\frac{\partial}{\partial x_i}\left(\rho\varepsilon u_i\right)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}\left(G_k+C_{3\varepsilon}G_b\right)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}$$

where, the eddy viscosity, μ_t, is written as,

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$$

The terms G_k and G_b represent the production of turbulence kinetic energy due to the mean velocity gradients and buoyancy respectively.

$$G_k = \mu_t S^2$$

$$G_b = \beta g_i \frac{\mu_t}{\mathrm{Pr}_t} \frac{\partial T}{\partial x_i}$$

where, S is the modulus of the mean rate-of-strain tensor, defined as,

$$S = \sqrt{2 S_{ij} S_{ij}}$$

The mean strain rate tensor, S_{ij}, is written as,

$$S_{ij} = \frac{1}{2}\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right)$$

The values used for the k-e model constants are: $C_{1\varepsilon}$ = 1.44, $C_{2\varepsilon}$ = 1.92, C_μ = 0.09, σ_k = 1.0, σ_ε = 1.3.

The radiative heat transfer was calculated using the discrete ordinates method. The discrete ordinates (DO) radiation model solves the radiative transfer equation for a finite number of discrete solid angles, each associated with a vector direction fixed in the global Cartesian system.

Turbulent combusting flows are characterized by continuous fluctuations in density, temperature, and species concentrations. Most fuels are fast burning, and the overall rate of reaction is controlled by turbulent mixing. In non-premixed flames, turbulence in the flow slowly mixes fuel and oxidizer into the reaction zones where they burn quickly. In premixed flames, the turbulence slowly mixes cold reactants and hot products into the reaction zones, where reaction occurs rapidly. In such cases, the combustion is said to be mixing-limited, and the complex, and often unknown, chemical kinetic rates can be safely neglected. Combustion proceeds whenever turbulence is present (i.e., $k/\varepsilon > 0$), and an ignition source is not required to initiate combustion. This is usually acceptable when modeling non-premixed flames. However, in premixed flames, the reactants will burn as soon as they enter the computational domain, ahead of the flame stabilization zone. To avoid this situation in combustion modeling, several approaches have been proposed to model the intricate turbulence-chemistry interactions in combustion applications. In this program, we used the Eddy Break-Up (EBU)/finite-rate formulation developed by Magnussen and Hjertager in 1976. Both the Arrhenius and turbulent mixing (or, eddy-dissipation) reaction rates are calculated. The net reaction rate is taken as the minimum of these two rates. In practice, the Arrhenius rate acts as a kinetic "switch'," preventing reaction before the flame stabilization point. Once the flame is ignited, the eddy-dissipation rate is generally smaller than the Arrhenius rate, and reactions are mixing-limited.

In turbulent flows, the species transport equation is written as:

$$\nabla \cdot \left(\rho \vec{v} Y_i\right) = -\nabla \cdot \vec{J}_i + R_i$$

where, the diffusive flux is

$$\vec{J}_i = -\left(\rho D_{i,m} + \frac{\mu_t}{Sc_t}\right)\nabla Y_i$$

Sc_t is the turbulent Schmidt number – the default value for which is 0.7. The term R_i in the species equation is the rate of production of species *i* due to chemical reaction. In the EBU-finite rate model, the calculation of the source terms in the species conservation equations was based on finite-rate chemical kinetics as well as the turbulent mixing of the species, in which both the eddy-dissipation (i.e., turbulent mixing) and Arrhenius reaction rates are calculated. The net reaction rate is taken as the minimum of these two rates.

In each control volume, the turbulent mixing rate is calculated as:

$$k_{mix} = A\frac{\varepsilon}{k}\rho \min\left(\frac{y_{fuel}}{M_{fuel}}, \frac{b}{a}\frac{y_{oxidizer}}{M_{oxidizer}}\right)$$

where, *A* is an empirical constant calibrated to be 4.0, and *b/a* is stoichiometric ratio between the fuel and oxidizer. For the calculation of the Arrhenius rates, two-step hydrocarbon oxidation reaction mechanisms, and a single-step hydrogen oxidation mechanism, were used.

$$CH_4 + 1.5O_2 \rightarrow CO + 2H_2O$$
$$C_3H_8 + 3.5O_2 \rightarrow 3CO + 4H_2O$$
$$CO + 0.5O_2 \rightarrow CO_2$$
$$H_2 + 0.5O_2 \rightarrow H_2O$$

with corresponding intrinsic reaction rate equations,

$$k_{f1} = 5.012 \times 10^{11} \exp\left(\frac{-2 \times 10^{8}}{RT}\right) C_{CH_4}^{0.7} C_{O_2}^{0.8}$$

$$k_{f2} = 5.62 \times 10^{9} \exp\left(\frac{-1.256 \times 10^{8}}{RT}\right) C_{C_3H_8}^{0.1} C_{O_2}^{1.65}$$

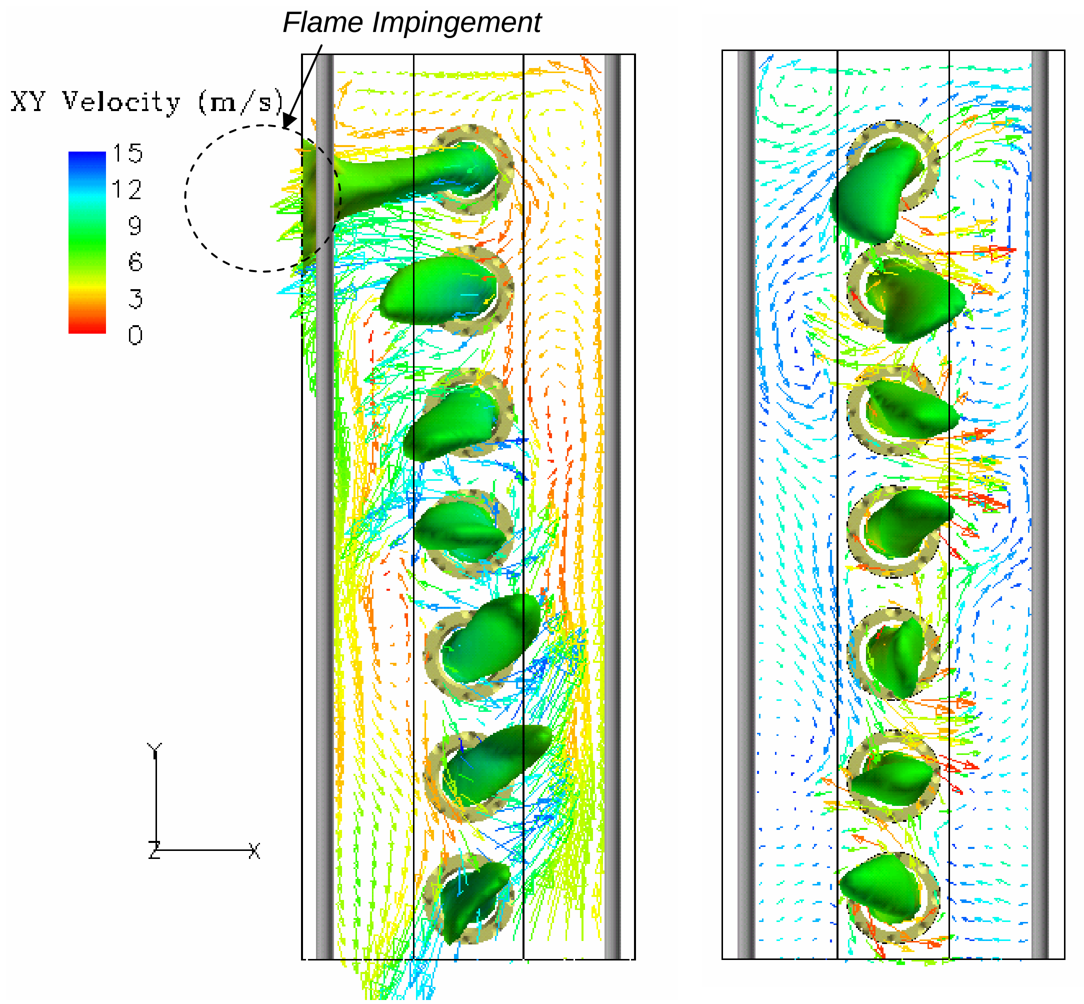
$$k_{f3} = 2.239 \times 10^{12} \exp\left(\frac{-1.7 \times 10^{8}}{RT}\right) C_{CO} C_{H_2O}^{0.5} C_{O_2}^{0.25}$$

$$k_{f4} = 9.87 \times 10^{8} \exp\left(\frac{-3.1 \times 10^{7}}{RT}\right) C_{H_2}^{1.0} C_{O_2}^{1.0}$$

Following the development of the model as described above, CFD modeling was used to optimize the first field application of the ultra low NO_x burner developed at TIAX and Callidus Technologies. The effectiveness of the ultra-low-emission burner was demonstrated in an atmospheric pipestill furnace at an ExxonMobil refinery. The demonstration furnace had a horizontal-tube, cabin configurations with a maximum firing rate of 140 MMBtu/hr. The furnace fired on fuel gas with compositions that varied from high methane to high hydrogen. A set of 14 field-test CUB-8 burners were installed for the retrofit demonstration replacing 18 original burners. A CFD analysis was performed for the heater radiant section prior to the retrofit to predict heater performance and identify potential problems (e.g., anomalous flow patterns or flame geometry).

Figure 3-9. The Demonstration Heater for the Ultra Low Emission Burner

The CFD results for the initial burner configuration with all co-rotating burners (i.e., same swirl direction in all burners) showed the formation of a large-scale rotation of the flow in the radiant box that deflected the flame adjacent to the end wall into the tubes (see Figure 3-10). Based on recommendations from TIAX, the burner layout was changed to a counter-rotating arrangement (i.e., opposite swirl directions in adjacent burners) that was predicted to eliminate the overall swirl pattern in the radiant box. The subsequent heater simulation with counter-rotating burners exhibited relatively straight flames.



(A) Co-Rotating Burners **(B) Counter-Rotating Burners**

Figure 3-10. CFD-based Optimization for the Ultra Low NO_x Burner Demonstration Heater

The demonstration heater trials were started in May 2001. During the trials, the flame geometry and flow patterns were found to be consistent with the CFD predictions, and the heat flux profile met ExxonMobil specifications. Burner stability was observed to be quite good when fuel composition was within specifications(Figure 3-11). However, some pulsation was experienced when methane content exceeded 85% by volume. Initial NO_x levels were also higher than anticipated at 0.030 lb/MBtu. Based on the results of these initial tests, a new flame stabilizer was developed for the CUB-8 burner and fuel gas tips were modified. These changes enhanced the burner stability and brought down the NO_x emissions to about 0.025 lb/MBtu.

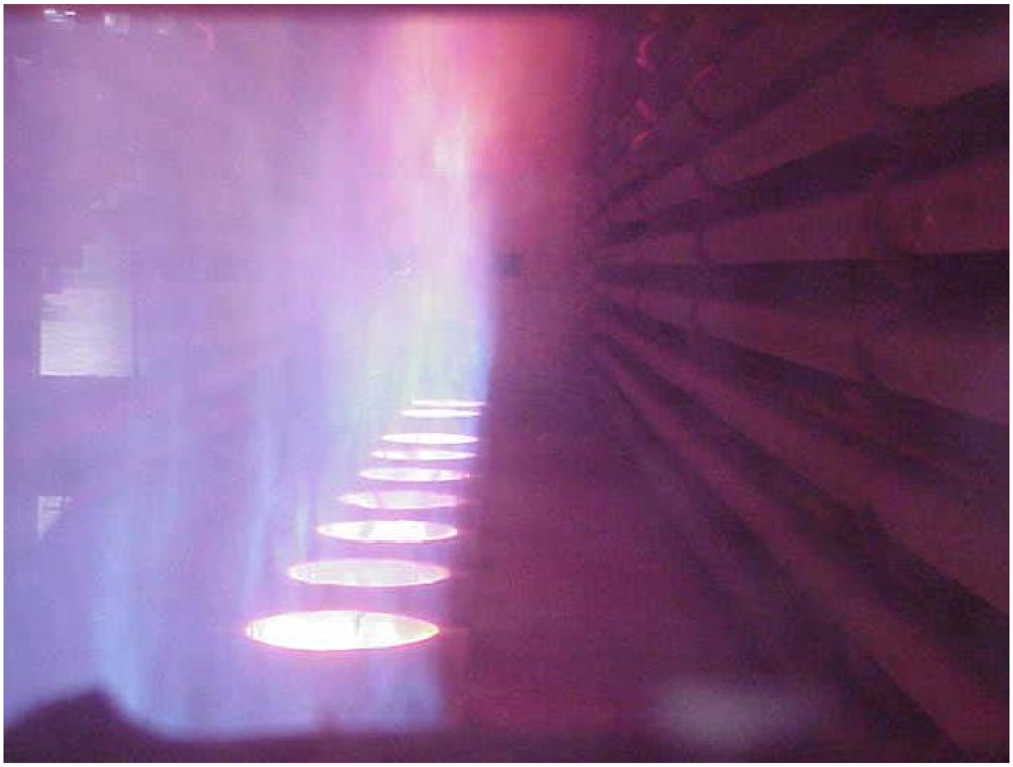


Figure 3-11. Flame Structure Observed in the Demonstration Heater after Retrofit with CUB-8 Burners

3.2 Advanced Fired Heater

3.2.1 Heater development

The integrated advanced fired heater (AFH) system design basis was a 100 MMBtu/hr vertical tube box heater operating with a thermal efficiency of 95% (LHV) fired with low sulfur refinery fuel gas and integrated with the ultra low NO_x burner to give lowest overall NO_x and greenhouse gas emissions. The AFH utilizes a unique combination of fired heater design concepts and heater components that can result in these high levels of efficiency and lowest possible emissions in a compact process fired heater unit.

The AFH design features that resulted from the design development sequence described in Section 2.4.2 are:

- Positive pressure design/operation avoids need for induced draft blower, minimizes air and flue gas ducting, and eliminates air-in-leakage thus results in lower cost, adds to improved thermal efficiency and lower NO_x emissions

- Radiant cells with a combination of one- and two-side fired radiant tubes results in reduced cost.

- Longitudinal finned convection section design incorporates high gas mass velocity thus minimizing effective heat transfer surface, resulting in lower cost and reduced external fouling which adds to improved thermal efficiency

- Integral, modular plant-type commercially available combustion air preheater

- Integrated with the commercially available CUB ULNB

Figure 3-12 shows the overall layout of the heater modules, air and flue gas handling sections and stack. Figure 3-13 shows detail of the radiative section, and Figure 3-14 shows the burner layout.

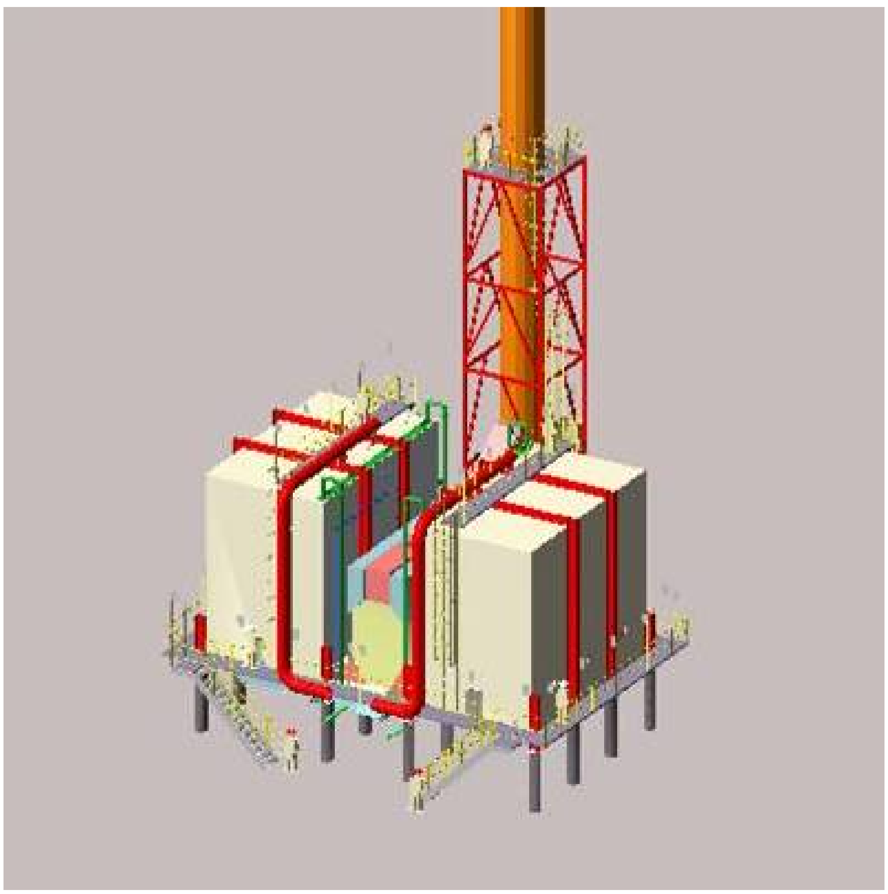


Figure 3-12. Advanced Fired Heater Configuration

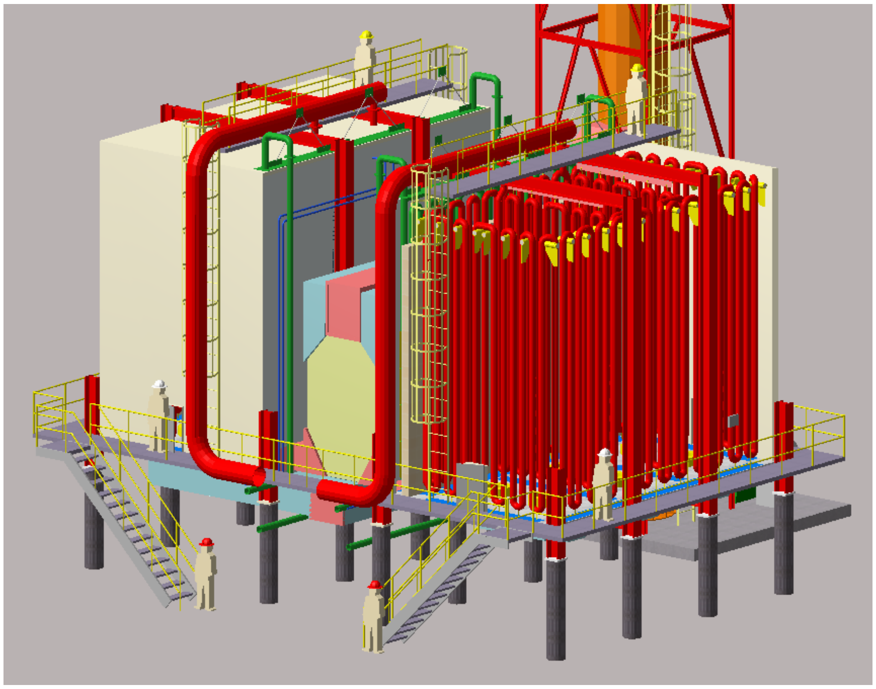


Figure 3-13. Advanced Fired Heater radiative section configuration

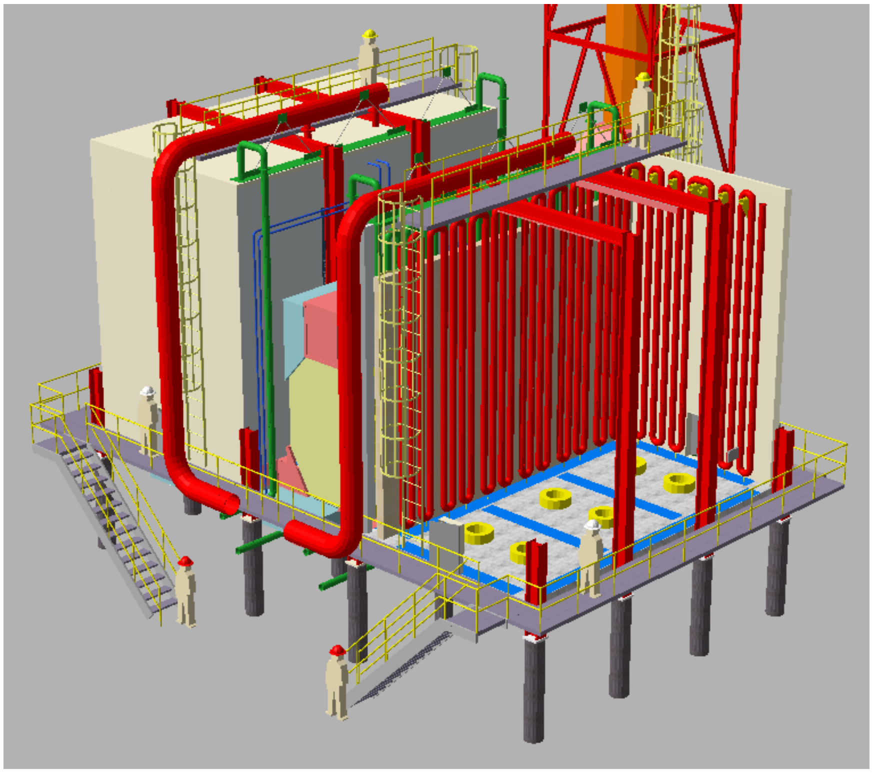


Figure 3-14. Advanced Fired Heater Burner Configuration

The optimized radiant section design utilizes concepts already commercially proven and in service in many applications. The convection section design, unlike conventional fired heaters is based on flue gases from the radiant section flowing axially downward along the length of the convection tubes. This convection design concept is unique in fired heater design applications. It utilizes the principles of double-pipe heat exchangers which is well proven in many other industrial applications. Flue gases from the convection section are ducted through a high efficiency air preheater to achieve the targeted heater efficiency while maximizing cold end metal temperature and thus minimizing condensation and fouling. The AFH design also utilizes optimum casing insulation to minimize system heat loss to maximize the heater efficiency. Design excess air levels for the AFH are lower than those typically used in conventional fired heaters. Extensive Computational Fluid Dynamics (CFD) modeling was utilized in determining the excess air level, coil layout, ducting, and sizing of various components leading to development of the AFH design.

Low emissions were a key design goal of the AFH development. The AFH design was based on using the Callidus Ultra Blue (CUB) burners in the CFD modeling. Burner-to-burner interactions were minimized in the radiant section geometry, thereby defining the size of the burner groups per zone. Burner-to-tube spacing criteria specified in API Standard 560 was followed in the AFH design. Required burner-to-burner spacing, between tile outside diameters, was maintained to facilitate flue gas re-entrainment into the flame zone and to minimize burner flame interactions.

The AFH design used CFD modeling extensively for thermal design of the radiant section, the convection section, and the flue gas and combustion air flow ducting, as discussed in Section 3.2.2.. The high efficiency combustion air preheaters and the forced draft blowers are standard equipment well proven in industrial applications. All other codes and standards that are typically applicable for thermal and mechanical design of a conventional fired heater, apply to the AFH design.

In addition to the enhanced efficiency and reduced emissions, the AFH design provides the following advantages relative to conventional designs:

- Operation at positive pressure, no air ingress
- Compact, modular unit with small footprint
- Low maintenance top supported tubes
- Two forced draft blowers and no induced draft blower

Table 3-1 summarizes projected costs, efficiency and emission performance for a 100 MMBtu/hr heater compared to conventional designs with and without combustion air preheaters. The benefits of the AFH design are:

- Lower total erected cost (10%)
- Lower NO_x, CO_2 emissions (3%)
- Lower fuel costs (3%)

Fuel savings from one installation of the AFH would be 131 billion Btu/yr compared to a conventional heater design and 26 billion Btu/yr compared to a conventional heater with air preheat. The low NO_x burner does not significantly affect heater efficiency.

Table 3-1. Estimated Performance of AFH

Heater Type	Efficiency % (LHV)	Heat Fired MMBtu/hr (LHV/HHV)	Total Erected Cost M$	NO_x Emissions lb/MMBtu (HHV) and tons/yr	CO_2 Emissions lb/MMBtu (HHV) and ktons/yr
Conventional fired heater	83	120/132	5.2	0.020/11.5	126/71.5
Conventional fired heater with air preheat	93	108/118	8.8	0.025/13	126/64.0
Advanced fired heater with air preheat	95	105/115	8.1	0.025/12.6	126/62.3

Capital cost savings = $700K (estimate performed July 2003), fuel cost savings based on $3.5/MMBtu = $92K/yr

Net present value of savings (15 yrs, 12% IR) > $1.0M per application

The advanced fired heater design has application to the following markets:

- Clean gas fired
- Non “heavy coking” i.e., vacuum heaters and coker heaters not suitable
- Absorbed duty greater than approximately 100 MMBtu/h
- Grass roots design applications, not suitable for retrofit

The AFH design is undergoing an extensive commercialization readiness review at EMRE. Approximately 90 percent of the steps in EMRE’s Technical Readiness Assessment Considerations (TRAC) have been successfully completed, and the AFH design is approaching the status of “technically ready”. Several key components, including the burners, radiant section, and combustion air preheater are all field proven. The convection section design has been proven in other applications. Thus, the design risks are considered much lower than a typical new design. A design manual has been prepared to support commercial deployment. Required steps for licensing the AFH technology are in progress.

3.2.2 Computational Fluid Dynamic Modeling

In 2001, prior to starting the design activities for the advanced fired heater (AFH), CFD modeling was used to understand the impact of using finned tubes in the radiant section of the AFH. Radiant tubes with fins on the back-side, facing the refractory, enhance heat pick-up on the back-side and reduce peripheral maldistribution. Moreover, the additional cooling of the flue gases flowing downward between the refractory and the tubes results in a significant temperature reduction of the flue gases near the furnace floor. This has a positive impact on the heater performance in the form of reduced NO_x emissions. Radiant tubes with axial fins on the tube back-side were modeled and compared with bare tubes. A single-burner test cell, shown in Figure 3-15, was used for this CFD-based analysis. The test cell had a height of 28 feet and a width of 9 feet. The cell was comprised of twelve 6-inch radiant tubes with a center-to-center spacing of 1 foot.

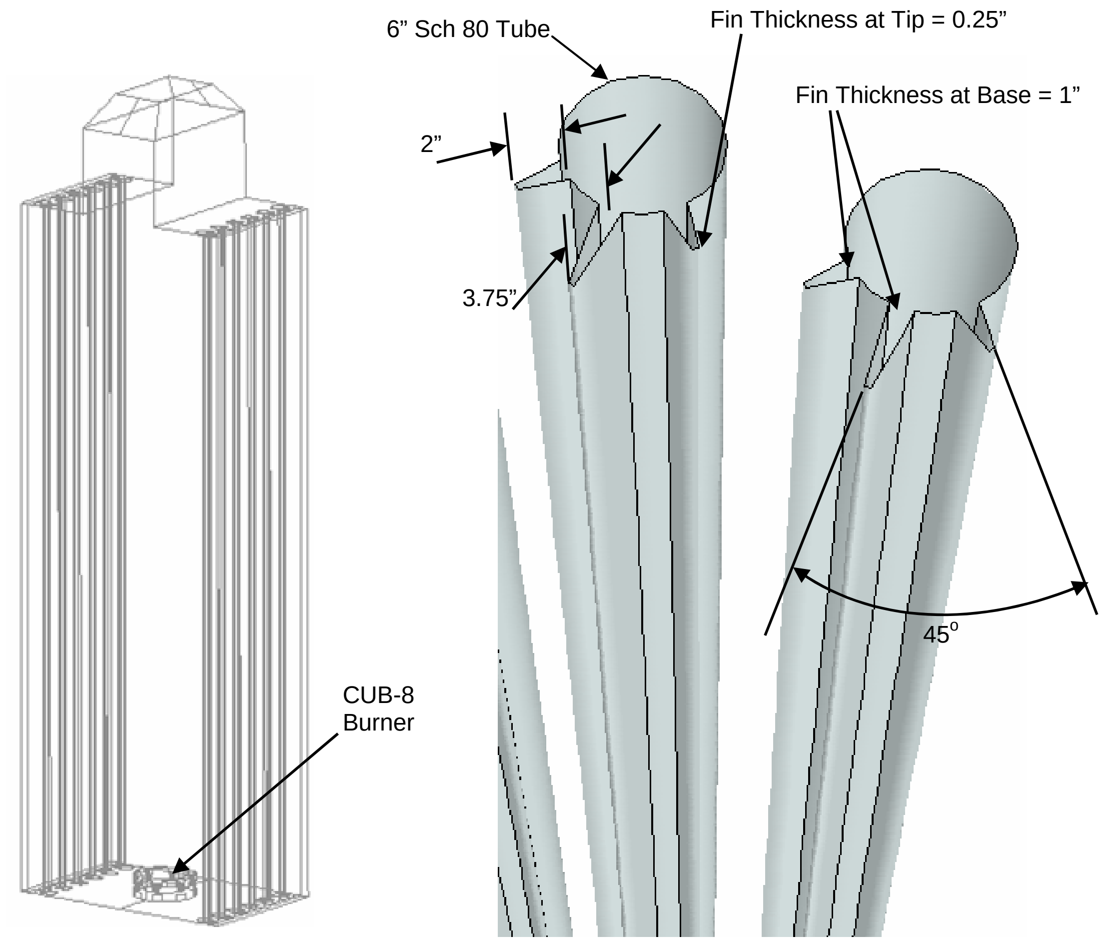


Figure 3-15. Single-Burner Test Cell Used to Evaluate Axially-finned Radiant Tubes with Bare Tubes

The burner used in this model was a Callidus CUB-8 burner with a firing rate of 9 MBtu/hr. For the process-side, a convective heat transfer boundary condition was used (average process fluid temperature = 745^{o}F, heat transfer coefficient = 140 Btu/hr-ft^2-R).

Although the CFD results showed almost similar heat transfer rates (about 3.2 MMBtu/hr) to the finned tubes and the bare tubes, the finned radiant tubes showed relatively lower peak heat flux values, and as a result, lower tube metal temperatures (see Figure 3-16). For the finned tubes, the lower temperature of the recirculating flue gases near the furnace floor resulted in reduced flame temperatures that brought the flame-side heat transfer down. However, the lower flame-side heat transfer for the finned tubes was offset by the additional heat pick-up on the tube back-side with final result of equivalent heat transfer in both the cases. Axially-finned radiant tubes, with their potential for lower tube metal temperatures, were therefore considered as a useful technology for the AFH.

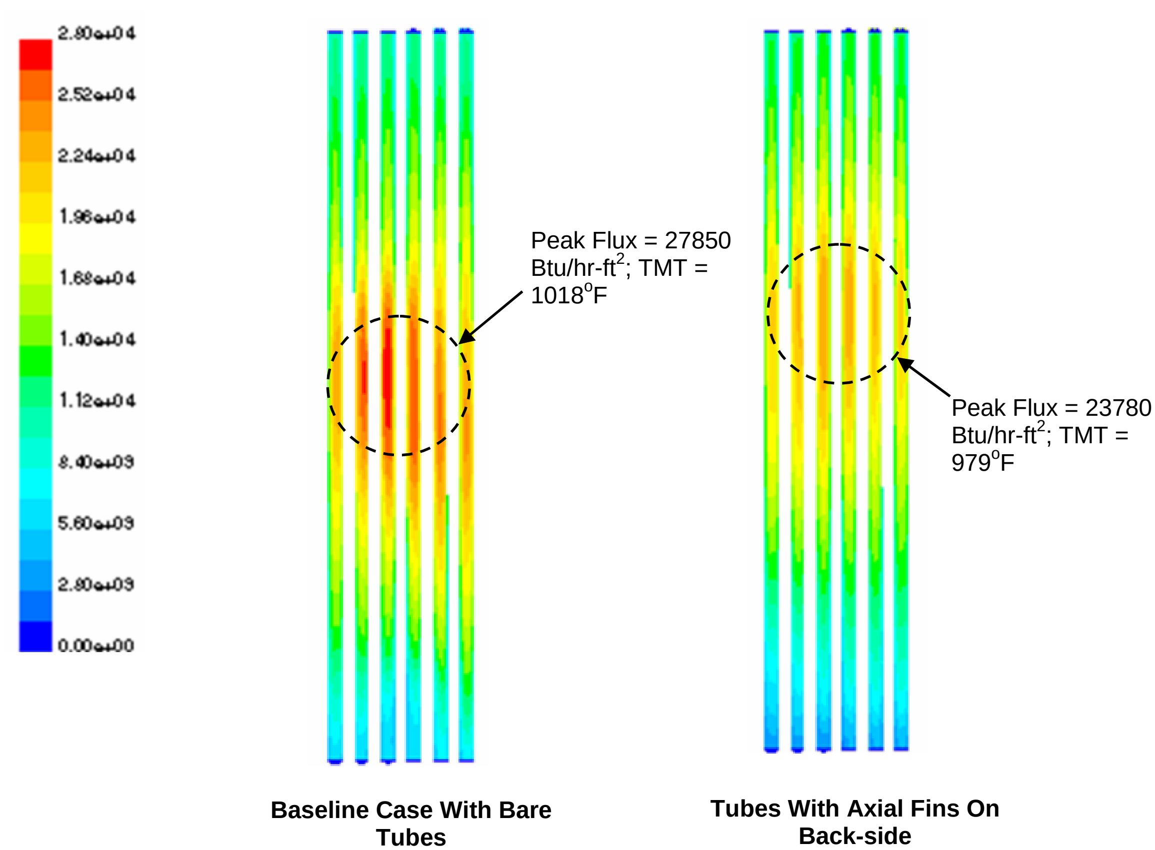


Figure 3-16. CFD results showing contours of surface heat flux on bare and finned radiant tubes in a single-burner test cell environment

The first design for the advanced fired heater was jointly developed by ExxonMobil Research and Engineering, Norton Engineering, and TIAX in early 2002. This heater design had a radiant box with a floor area of 31' x 26'9", and a height of 26'-6". The radiant box was further divided into three radiant cells, each with four CUB-8 burners. The burner center-to-center spacing was kept at 7'-2". The single-fired tubes adjacent to the radiant box walls were designed with axial fins on the back-side facing the refractory (the fin dimensions were kept the same as those shown in Figure 3-17). The double-fired tubes separating the radiant cells were kept bare. Unlike conventional fired heaters, the convection tubes were not placed above the radiant box. The convection tubes in this case were finned, vertical tubes suspended from the furnace roof along two sides of the heater. A 23-foot high brick wall was used to separate the convection section tubes from the radiant box. In other words, the flue gases rising up from the CUB burners would have to make a 180° turn just above the brick wall in order to enter the finned convection section.

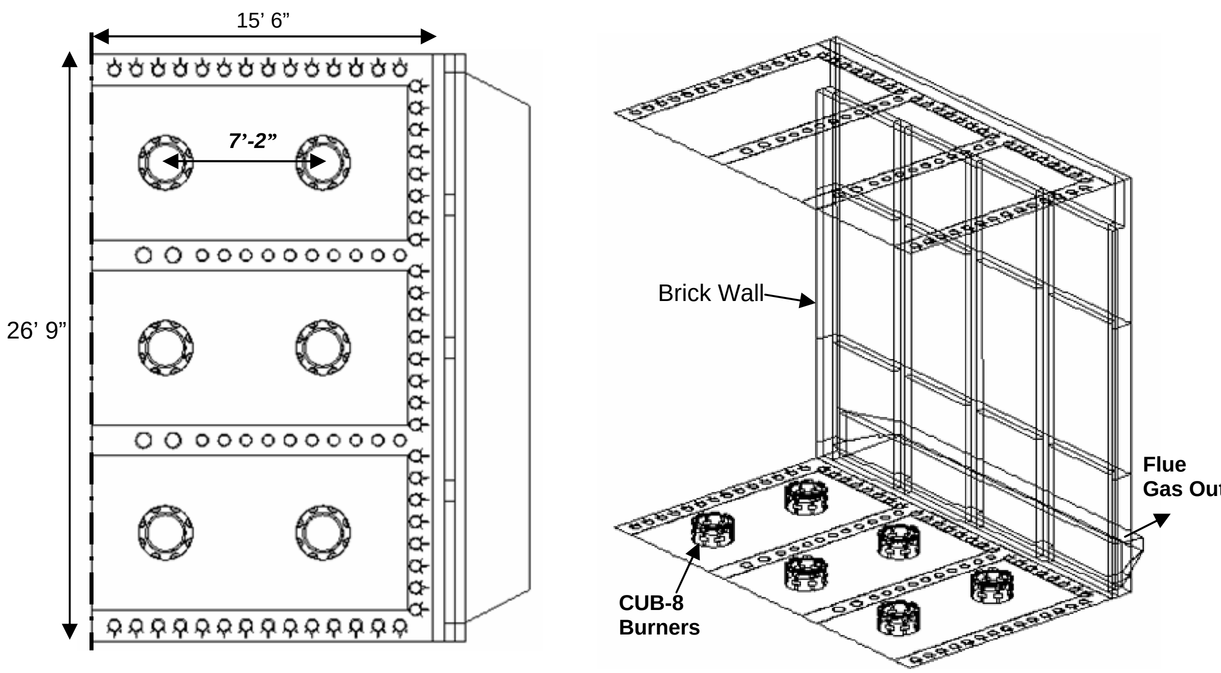


Figure 3-17. AFH Design Developed in Early-2002 (half-heater model shown above)

As described earlier, a detailed single-burner CUB-8 model was first run to generate the boundary conditions for the multi-burner AFH CFD model (Figure 3-18). The single CUB-8 burner was modeled with a firing rate of 8.75 MMBtu/hr (LHV) per burner and 666^{o}F preheated combustion air. Methane was used as the fuel for these simulations.

Data was extracted from a plane inside the burner tile near the burner exit. The information transferred to the AFH model included velocity components, temperature, species mass fractions, and turbulence quantities (such as the turbulent kinetic energy and the turbulence dissipation rate). The AFH model was run with the imported boundary condition data from the single-burner model and burners with alternating swirl directions in individual radiant cells. The radiant tubes were modeled with the tube metal and fin thicknesses. The process-side was modeled using convective boundary conditions – obtained from ExxonMobil – on the inside surface of the radiant tubes (heat transfer coefficient of 140 Btu/hr-ft^2-F; bulk process fluid temperature of 745^{o}F).

The CFD results were found to be quite acceptable (see Figures 3-19, 3-20). The primary objective of the analysis was to identify undesirable flow patterns in the heater, and evaluate corrective modifications, if necessary. The predicted flame structure showed relatively straight flames, and a total of 83.6 MMBtu/hr heat absorbed in the entire radiant box (with 12 burners). Nonetheless, the peak heat flux value for the 2-side fired tubes (>30,000 Btu/hr-ft^2) was of some concern. It was felt that the close spacing between the burners and the double fired tubes caused the high heat flux levels on these tubes.

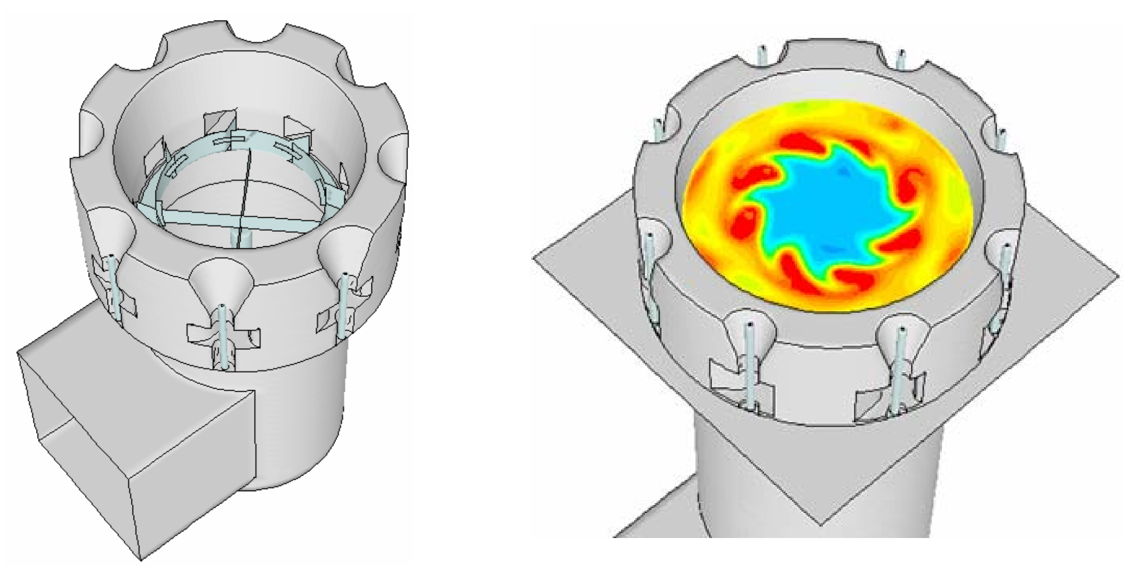


Figure 3-18. Single-burner Model Geometry, Along with Temperature Contours in a Plane Used to Transfer Between the Single-burner Model and the AFH Model

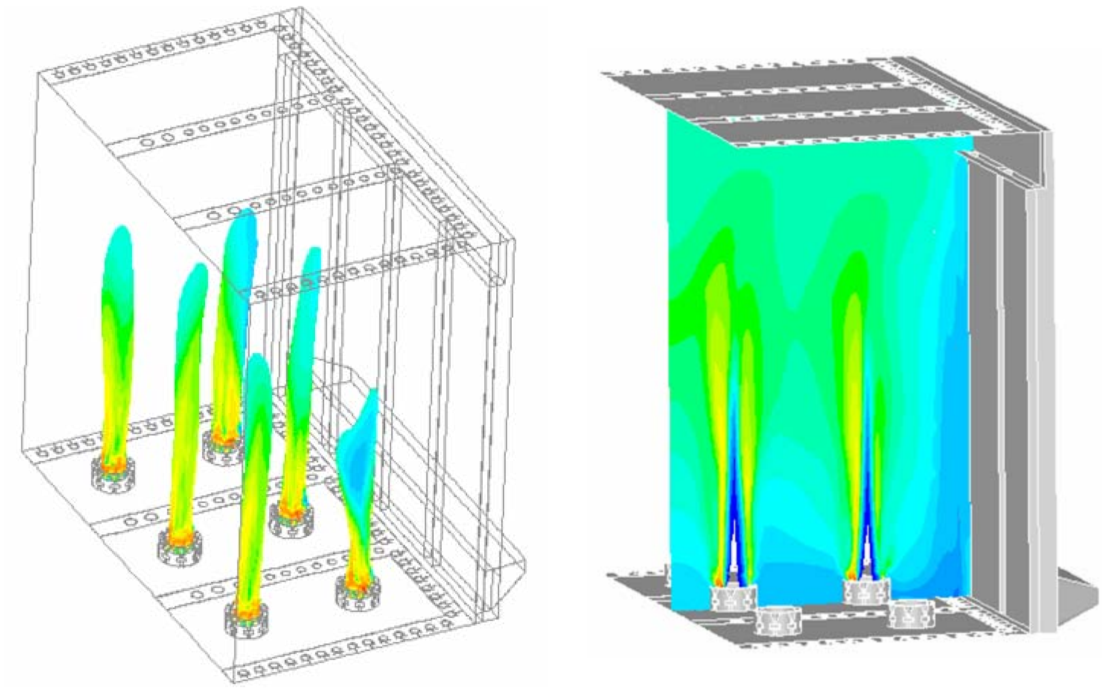


Figure 3-19. CFD solution for the AFH with counter-swirl burners. Flame structure (left) represented by iso-surfaces of 6.5% oxygen, and contours of temperature (right)

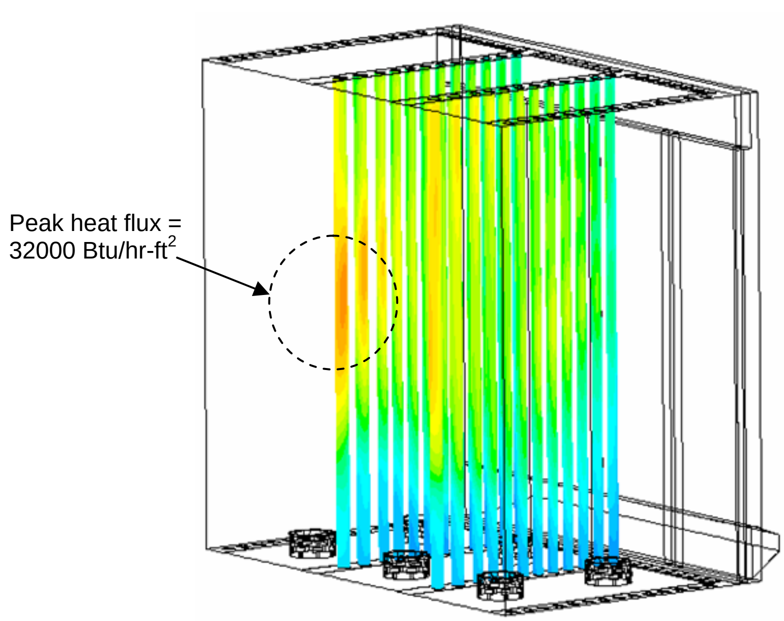


Figure 3-20. Heat flux values plotted on the AFH double-fired tubes

A subsequent CFD analysis of this AFH design with co-swirl burners (i.e., all burners in a radiant cell had the same swirl direction) was performed in an effort to reduce the flux levels on the 2-side fired tubes. This analysis, however, indicated the possibility of flame impingement on the tubes, and was not considered for further evaluation (see Figure 3-21).

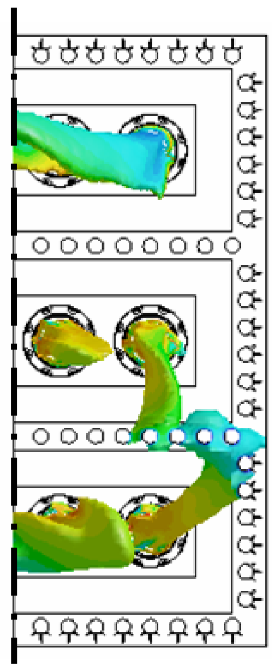


Figure 3-21. CFD solution for the AFH with co-swirl burners. Flame structure represented by iso-surfaces of 6.5% oxygen

After the reasonably successful computer-based demonstration for the viability of the initial AFH design, the focus shifted to reducing the flux levels on the double-fired tubes and having a common plenum for the flue gases. During late-2002, a second-generation AFH design was created by ExxonMobil Research and Engineering and TIAX.

This design increased the heater width from 26'9" to 28'3" – resulting in more distance between the burners and the double fired tubes – and reduced the radiant box length from 31' to 28'10". The latter was brought about by bringing the burners closer (from 7'2" to 4'5"). Also, the radiant box height was reduced to 25'. The new design incorporated six radiant cells with two counter-swirl CUB-8 burners in each cell. The convection tubes were relocated to the heater center that facilitated the collection of flue gases from all radiant cells in a common central plenum below the heater floor. The single-fired tubes were designed with axial fins on the back-side, while the double fired tubes were, as in the previous design, kept bare. The overall AFH design is shown in Figure 3-22. This design was thoroughly evaluated using CFD techniques as well. The approach remained the same – specific data (temperature, species mass fractions, etc.) from the single-burner model were imported into the new AFH model, and the model was run until sufficient convergence was achieved.

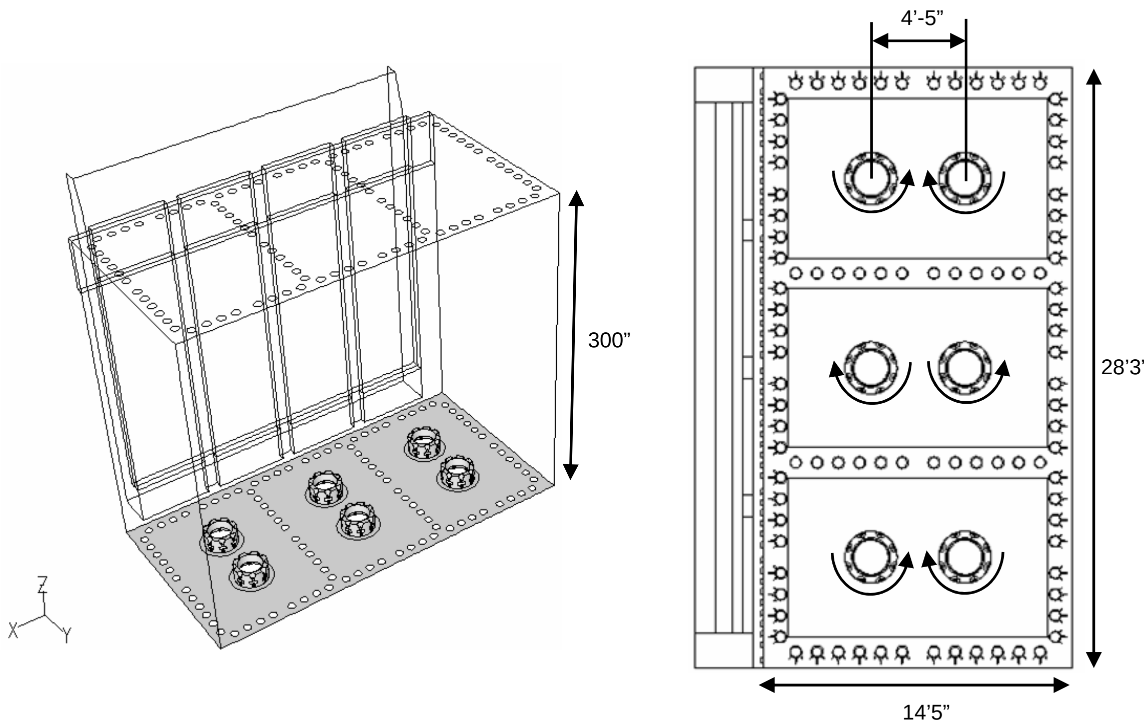
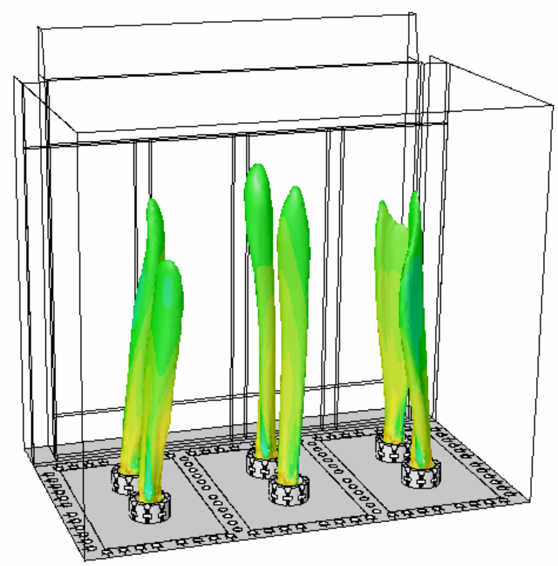


Figure 3-22. AFH design developed in late-2002 (half-heater model shown above)

The CFD results for the new AFH design were quite encouraging. The flame-structure was characterized by relatively straight flames – although the flames in the side cells showed a slight lean. The peak heat flux levels on the double fired tubes exhibited a reasonable decrease to 27,000 Btu/hr-ft^2 (Figure 3-23). And, the overall heat transfer in the radiant box remained about the same at 82.6 MMBtu/hr (for the entire 12-burner AFH).



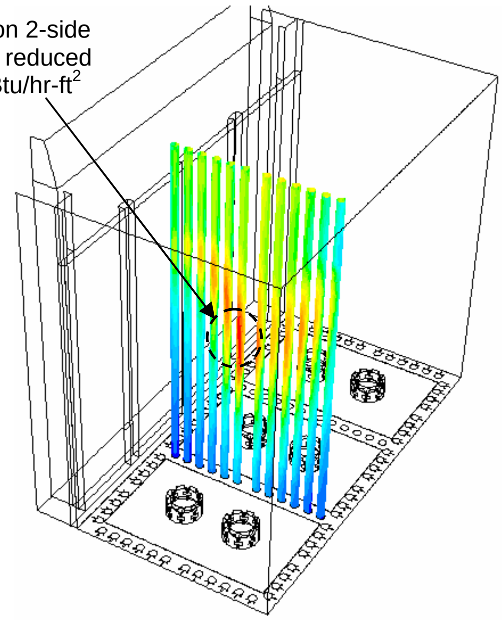


Figure 3-23. CFD results for the second-generation AFH design

At this point, after the successful results from the CFD modeling, manufacturing considerations were appraised for the AFH. Unfortunately, it was found that the axially-finned tubes required for the AFH design were not quite feasible in terms of manufacturing because of the long weld lengths. In fact, studs (or, cylindrical spines) – which could be easily resistance welded to the tubes – were determined to be a better alternative to axial fins.

However, the studs were found to be computationally hard to evaluate. Given the complex geometry associated with the studs, the number of numerical finite volumes required per radiant tube increased exponentially. In short, it was no longer possible to run an AFH model consisting of three radiant cells with studded tubes. It was decided to evaluate studded tubes using a CFD model for a single radiant cell. The studded radiant tube geometry was developed with studs on the rear-most 90^o facing the refractory as shown in Figures 3-24 and 3-25. The 2-inch long, 0.5-inch diameter studs were arranged in a staggered configuration to maximize the heat transfer. The distance between consecutive stud rows was kept at 7/8". Two studded tube configurations were developed and evaluated using CFD – (1) a fully-studded tube with studs along 18 feet on the tube back-side, and (2) a half-studded tube with studs along 9 feet on the tube back-side (Figures 3-26, 3-27). AFH single-cell models were built with these studded tubes. Additionally, a bare-tube model was also developed for the purpose of performance comparison.

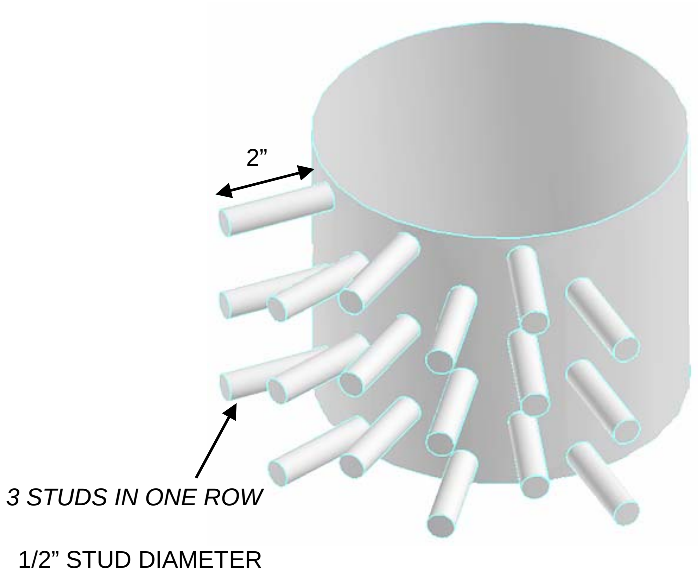


Figure 3-24. Stud geometry used in the AFH simulations

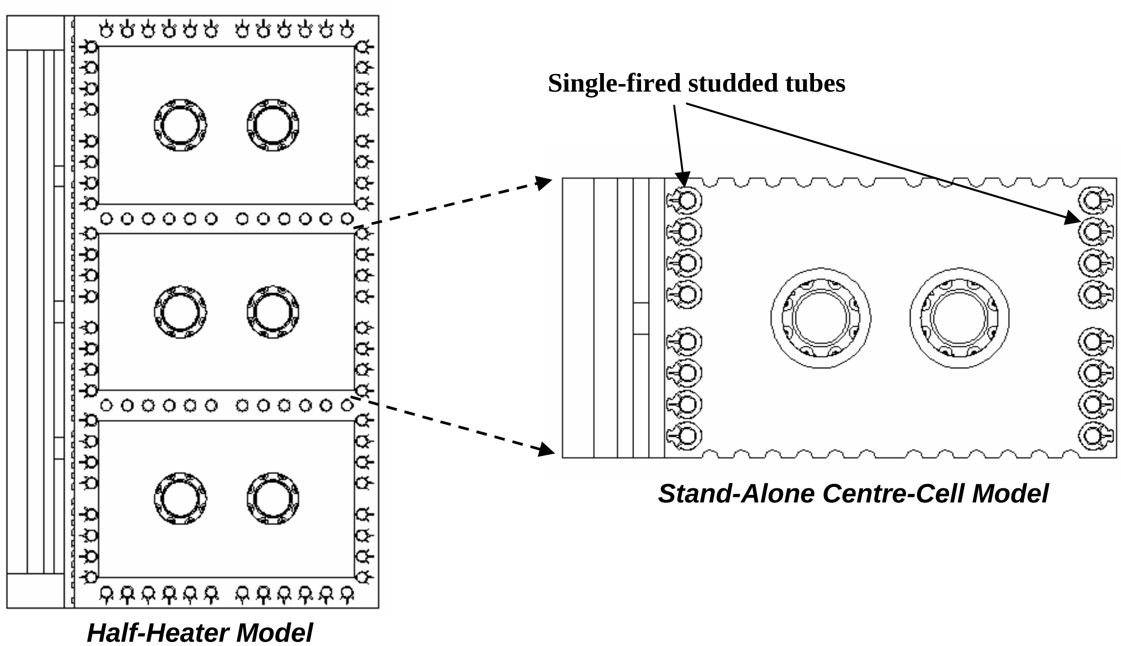


Figure 3-25. Single-cell model used for the evaluation of studded tubes in the AFH

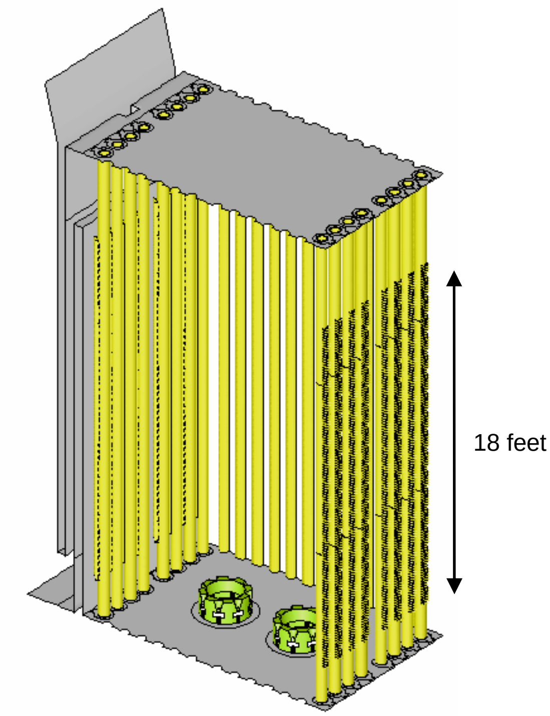


Figure 3-26. AFH model with fully-studded tubes

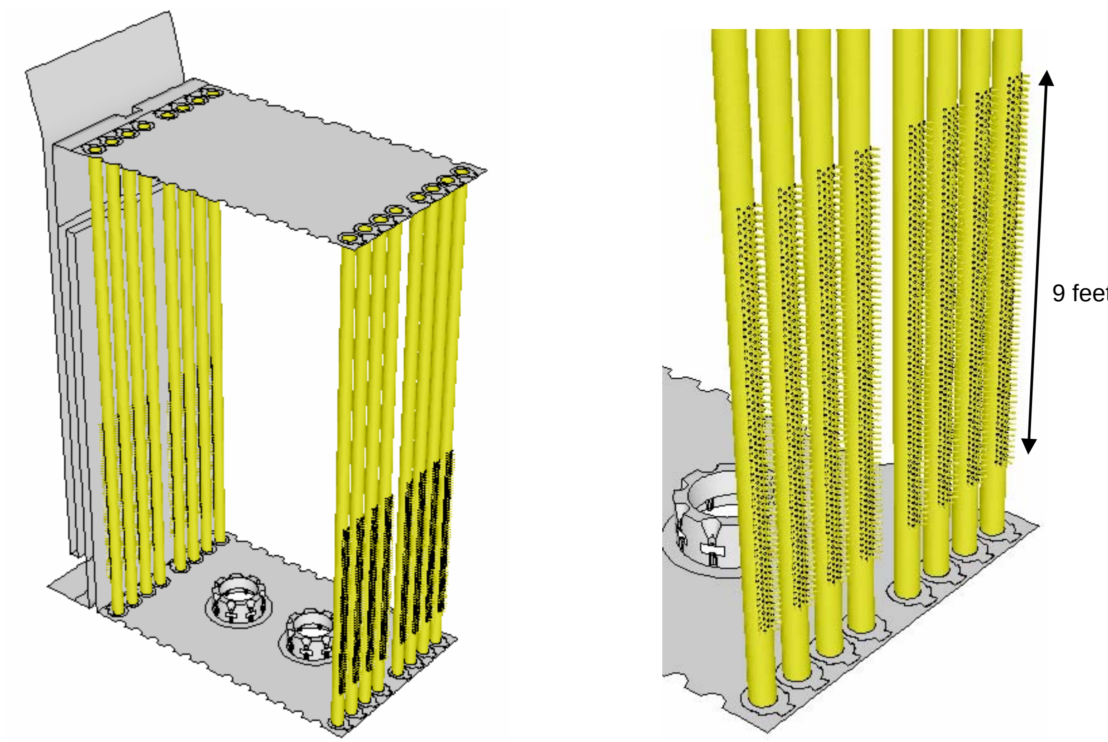


Figure 3-27. AFH model with half-studded tubes

The CFD analyses performed for the three single-cell models did not bring forth any clear advantages in using studded tubes. The CFD run with the fully-studded tubes showed the formation of a large flue gas recirculation zone created by the enhanced cooling on the back-side of the studded tubes and the resulting buoyancy-induced downdraft. The impact of this recirculation was a tilt in the flame from the outer burner in the direction of the double-fired tubes (see Figure 3-28). The net effect was a high peak heat flux value for the double-fired tubes, and a heat flux maldistribution that reduced the radiant box efficiency when compared to the bare tube results, Figure 3-29. The only advantage that the model predicted in using fully-studded tubes was the slightly lower flue gas temperature near the furnace floor, suggesting the possibility of somewhat lower NO_x emissions.

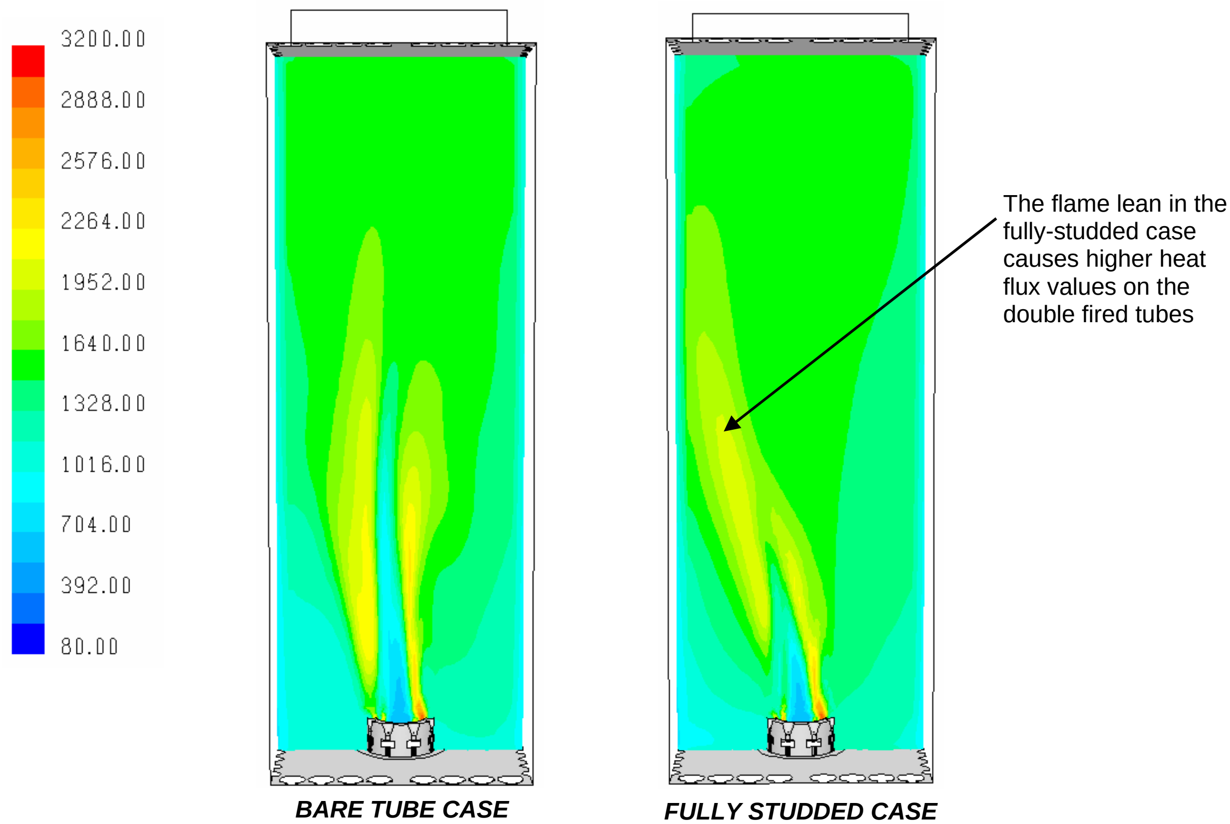


Figure 3-28. Contours of temperature for the bare-tube model and the fully-studded tube model showing a flame lean in the fully-studded case

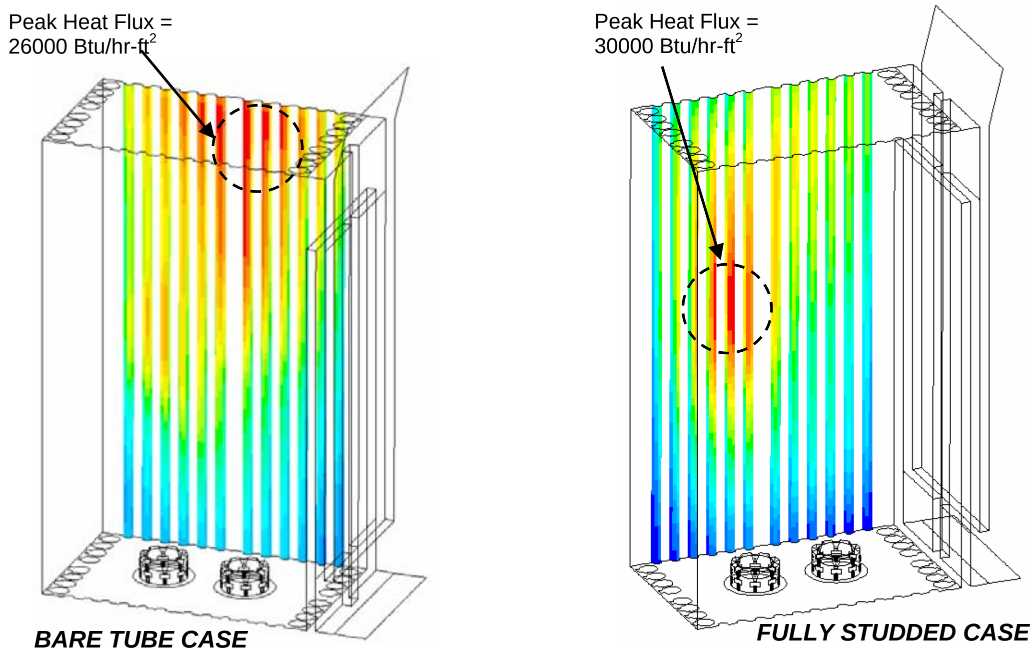


Figure 3-29. Contours of heat flux on double-fired tubes showing a higher heat flux in the fully-studded tube model caused by a flame lean

The half-studded tubes fared no better compared to the fully-studded tubes. Although the radiant box efficiency climbed back to a value similar to that from the bare tube model, the half-studded model also exhibited higher furnace-floor temperatures and unacceptable stud tip temperatures (1245^{o}F, a value unacceptable for carbon steel studs)(Figure 3-30). This was likely due to the shorter, hotter recirculation region created by the half-studded tubes. In sum, neither the fully-studded model nor the half-studded model predicted any results that justified the additional cost of the studded tubes. On the other hand, the radiant box model with just bare tubes – something not considered in the first-generation and the second-generation AFH designs – showed better performance. All along it had been assumed that finned radiant tubes would perform better than bare ones based on the 2001 single-burner CFD model. The subsequent models revealed that the performance of a heater is sensitive to the flame interaction in a specific heater configuration. In this case, although the studs did enhance heat transfer in the radiant section, the side-effects in the form of undesirable flow patterns that caused reduced thermal efficiency, higher floor temperatures, or elevated metal temperatures, made them impractical for the AFH. Results are summarized in Table 3-2.

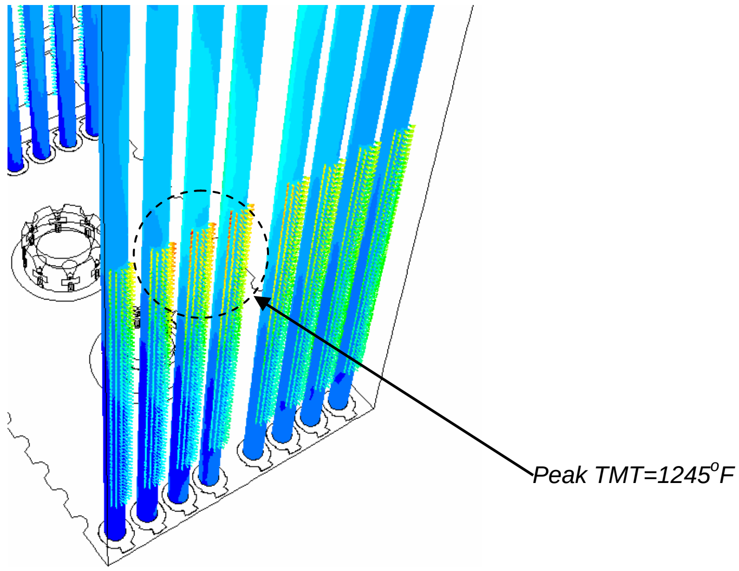


Figure 3-30. Contours of metal temperature on the half-studded tubes showing unacceptable stud tip temperatures

Table 3-2. Summary of CFD results for the single-cell AFH model with bare, half-studded, and fully-studded tubes

	Bare Tubes	**Half-Studded Tubes**	**Fully-Studded Tubes**
Heat absorbed by single-fired tubes	6.6 MMBtu/hr	6.6 MMBtu/hr	6.2 MMBtu/hr
Percentage of total heat absorbed by studs in single-fired tubes	N/A	3.6%	7.6%
Heat absorbed by double-fired tubes	6.7 MMBtu/hr	6.7 MMBtu/hr	6.6 MMBtu/hr
Total heat absorbed in radiant section	13.3 MMBtu/hr	13.3 MMBtu/hr	12.7 MMBtu/hr
Average temperature of flue gas near the burners	1180°F	1221°F	1152°F
Peak stud tip temperature	N/A	1245°F	1180°F

Towards the end of 2005, it was decided by ExxonMobil and TIAX to further evaluate the bare single-fired radiant tubes for the AFH design. A new three-cell (or half-heater) AFH radiant box model was developed with all bare tubes for direct comparison with the previous three-cell model with axially-finned, single-fired tubes. The results showed a flame structure similar to that observed in the model with the axially-finned tubes – straight centre-cell flames, and slightly leaning end-cell flames as a result of the end-wall recirculation (Figures 3-31, 3-32). The bare-tube model was characterized, however, by better radiant box efficiency, slightly higher floor temperatures, and slightly higher flux levels on the double fired tubes (Table 3-3).

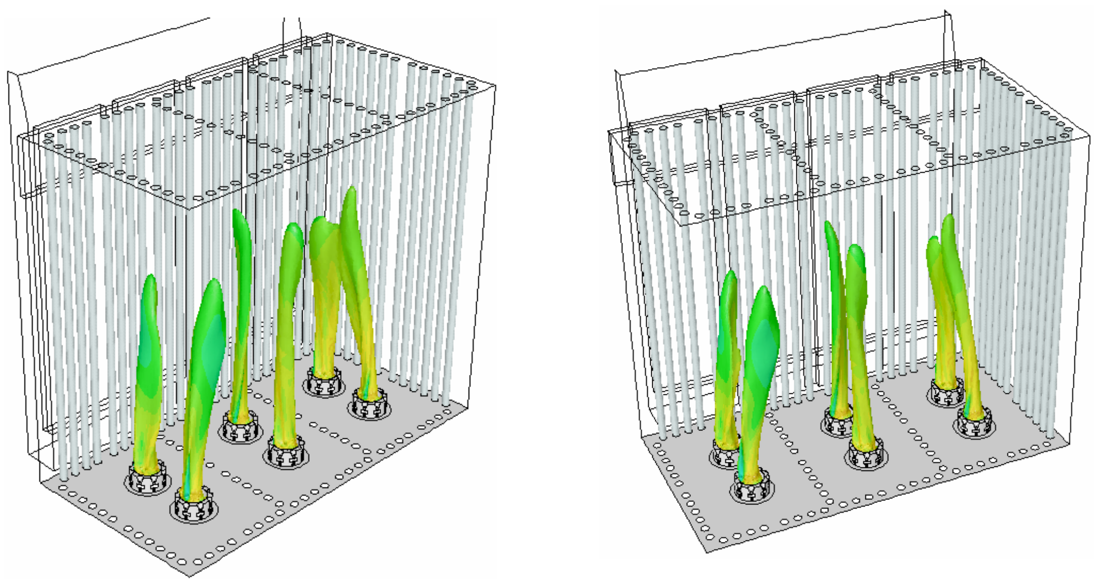


Figure 3-31 Flame structure prediction for an AFH design with all bare tubes

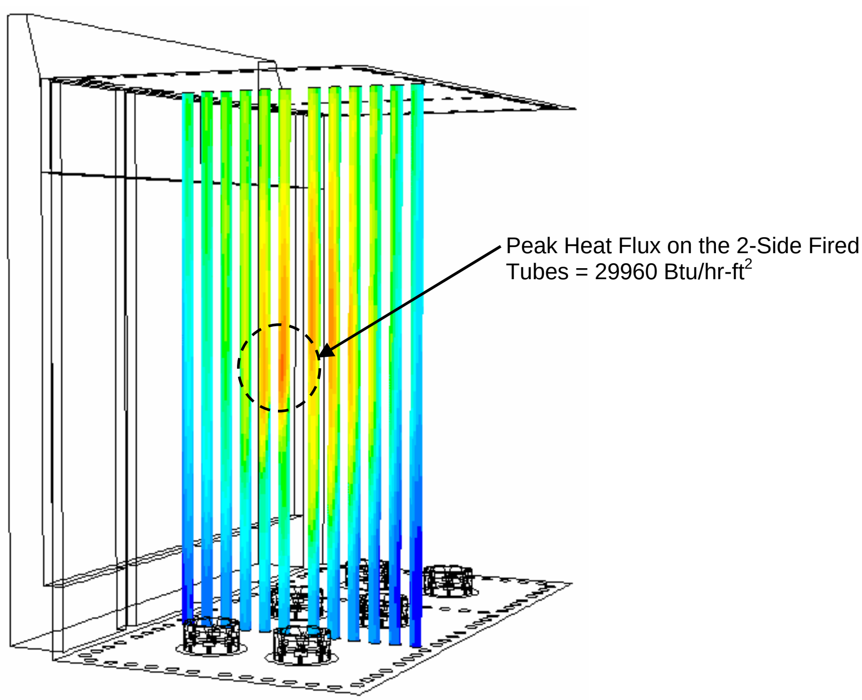


Figure 3-32. Flux contours on double-fired tubes in an AFH design with all bare tubes

Table 3-3. Comparison of the bare-tube AFH radiant box design with the previous AFH design with axially-finned, single-fired tubes on a full heater basis

	Bare Tubes	Axially-Finned Tubes
Total heat absorbed in radiant box	84.2 MMBtu/hr	82.6 MMBtu/hr
Peak heat flux on single-fired tubes	25900 Btu/hr-ft^2	23500 Btu/hr-ft^2
Peak heat flux on double-fired tubes	30000 Btu/hr-ft^2	27400 Btu/hr-ft^2
Average flue gas temperature near the burners	1100^{o}F	1050^{o}F

The CFD model results for the AFH design with bare tubes were found to be acceptable. In early 2006, a final AFH model was developed and run as shown on Figure 3-33. This model served two purposes. First, throughout the CFD analyses for the various AFH designs, the radiant tubes in the models had a length equal to the box height that made the meshing considerably easier. This was corrected in this final model; radiant tubes with an effective length of 22 feet were built into the model. Second, the convection section tubes suspended from the heater roof have a bare portion of about 3 feet exposed to the radiant box. It turned out to be difficult to estimate the radiative heat flux on these tubes. In the final AFH model, the exposed portion of the convection section tubes was incorporated in the model. In other words, we used the radiative heat transfer model in the CFD code to calculate the radiative heat transfer to the convection tubes. Also, the tube material was changed to SA-213 T9 steel from carbon steel in order to withstand the relatively higher flux levels observed in the previous bare-tube model.

The CFD results showed a flame pattern almost similar to that seen in the previous AFH model (Figures 3-34 through 3-37, Table 3-4). The overall radiant box duty reduced to 80.8 MMBtu/hr as expected due to the shorter radiant tubes. The temperature field near the burners – which is the temperature of entrained flue gases into the low-NO_x CUB burners – went up marginally to 1150^{o}F. The peak heat flux levels in the radiant box remained below 30000 Btu/hr-ft^2. The peak flux value on the double fired tubes translated into a tube metal temperature of 1070^{o}F (calculated outside of the CFD model using the estimated local process fluid temperature). All these values were found to be acceptable for the AFH design. In addition, the heat absorbed by the bare portion of the radiant tubes was calculated as 4.3 MMBtu/hr, which was higher than that estimated by hand calculations. This was largely because the hand calculations underestimated the radiative heat transfer to this portion of the convection tubes, calculated to be almost 85% of the total heat transfer from the CFD model.

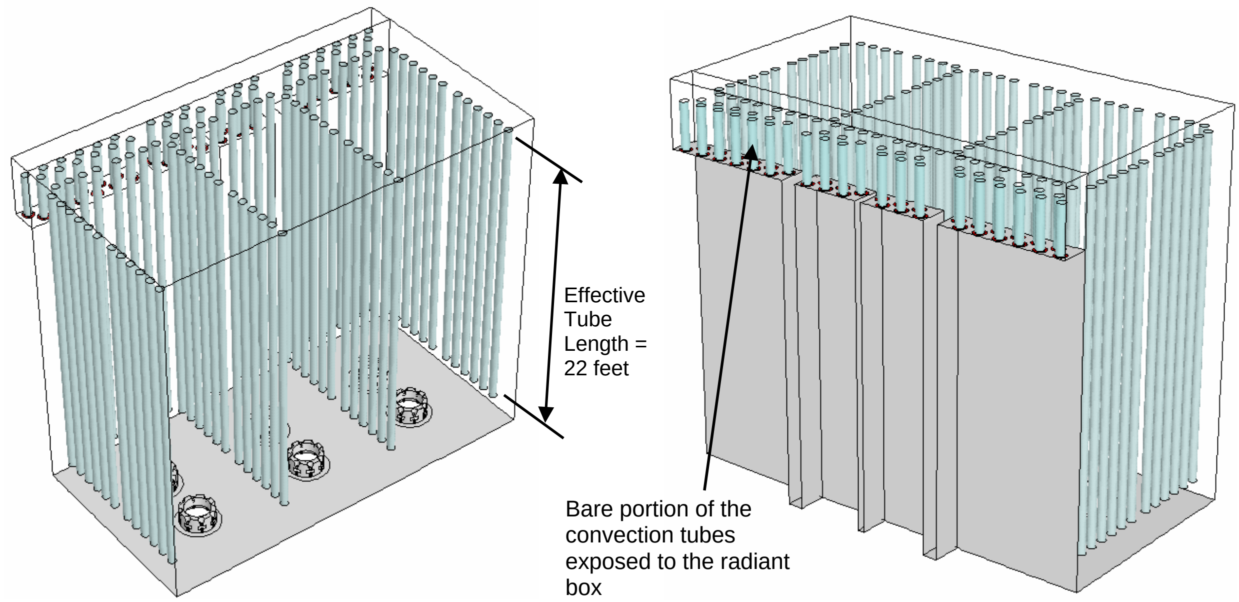


Figure 3-33. Final AFH model with the effective radiant tube length and the bare portion of the convection section tubes

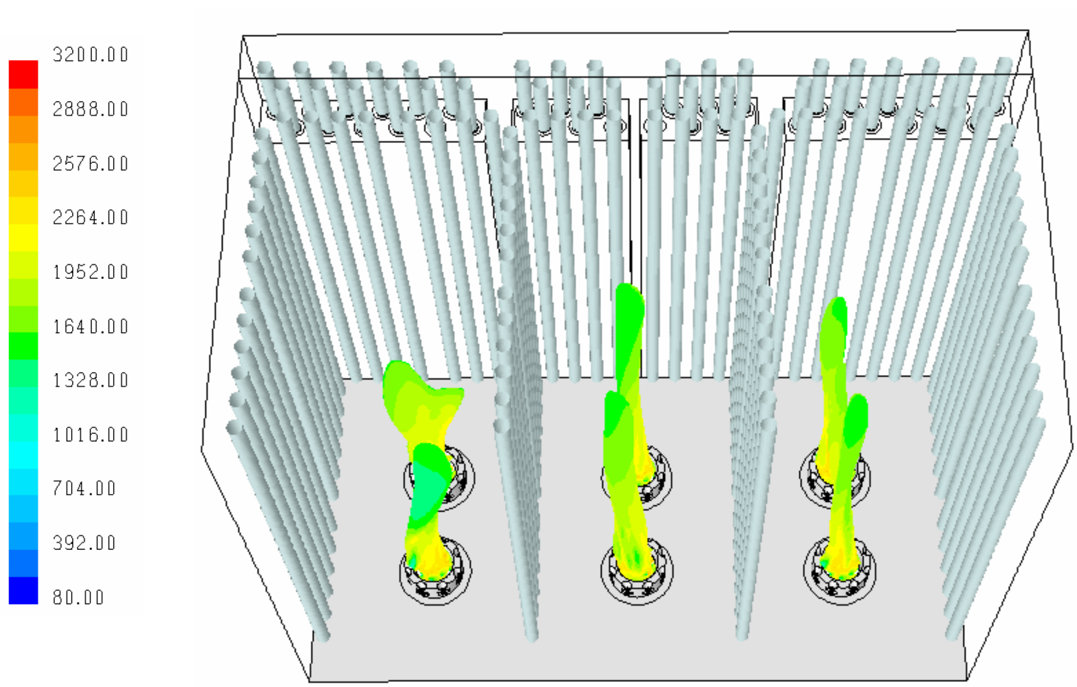


Figure 3-34. Predicted Flame Structure in the Final AFH model

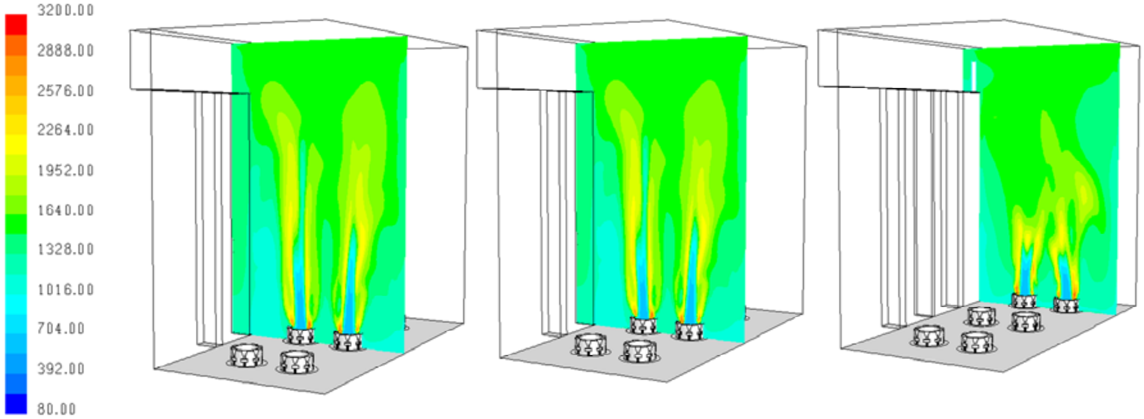


Figure 3-35. Contours of temperature through all the burners modeled in the Final AFH model

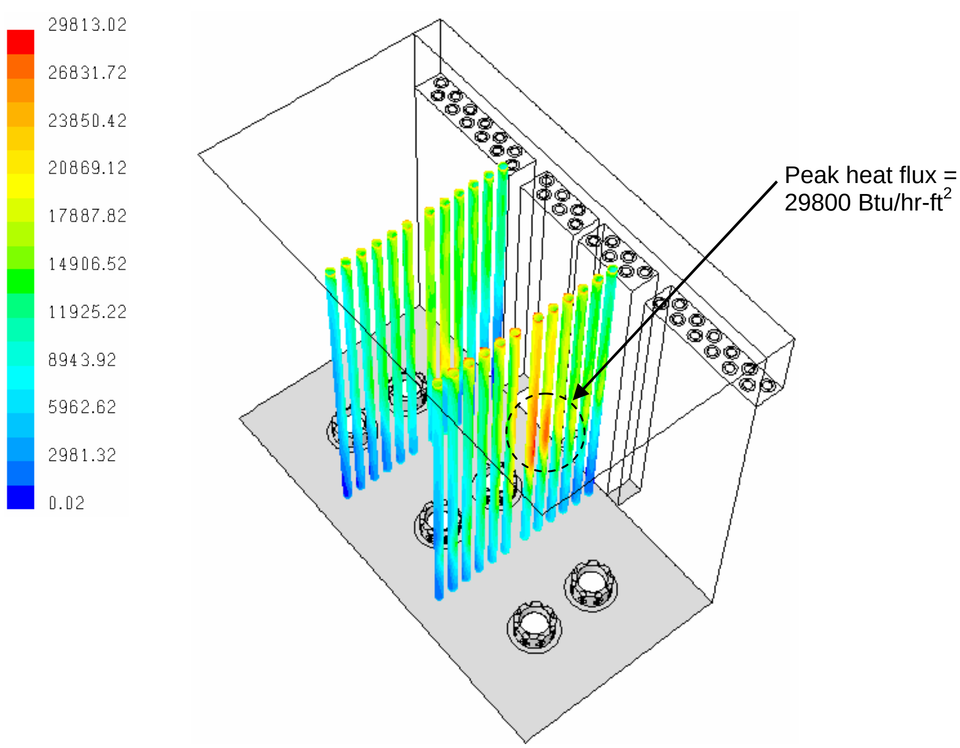


Figure 3-36. Contours of heat flux on double-fired tubes in the Final AFH model

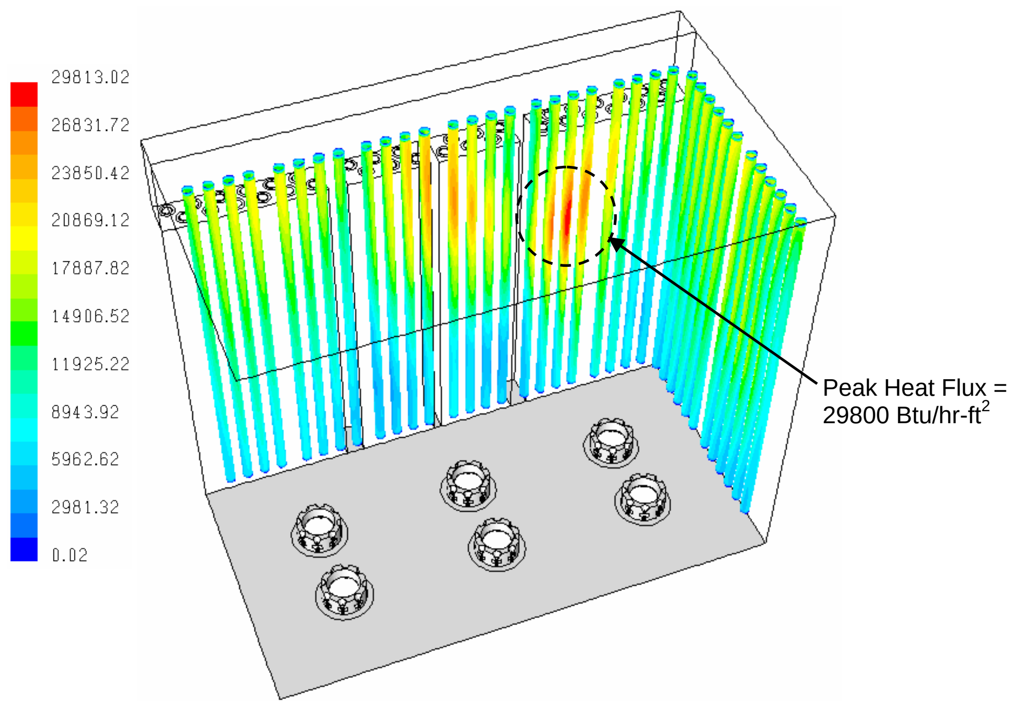


Figure 3-37. Contours of heat flux on single-fired tubes in the Final AFH model

Table 3-4. Summary of CFD Results for the Final AFH Model on a Full Heater Basis

Parameter	Value
Heat Absorbed by single-fired tubes	57.0 MMBtu/hr
Heat Absorbed by double-fired tubes	23.8 MMBtu/hr
Total Heat Absorbed in Radiant Box (% Radiative Heat Transfer)	80.8 MMBtu/hr (89%)
Peak Heat Flux in Radiant Box	29800 Btu/hr-ft^2
Average Flue Gas Temperature near the Burners	1150^{o}F
Heat Absorbed by the bare portion of Convection Tubes (% Radiative Heat Transfer)	4.3 MMBtu/hr (85%)
Flue Gas Temperature at the Inlet to the Finned Portion of the Convection Section	1258^{o}F

Flow and Heat Transfer Modeling for the AFH Convective Section

The AFH radiant box heat duty was calculated to be 80.8 MMBtu/hr in the final AFH model. Further, the bare portion of the convection section tubes accounted for a heat transfer of another 4.3 MMBtu/hr to the process fluid. Since the goal of the program is an overall thermal efficiency of 95%, the total target duty for the AFH is 99.75 MMBtu/hr with the design firing rate of 105 MMBtu/hr. In other words, the finned portion of the convection tubes would need to have a duty of about 14.6 MMBtu/hr to meet the program efficiency targets. The finned convection tubes were, as before, jointly designed by ExxonMobil Research and Engineering, TIAX, and Norton Engineering. The convection section was designed with 72 6-inch schedule-80 convection tubes suspended from the heater roof. The convection section tubes were provided with 0.05-inch thick axial fins to enhance the heat transfer to these tubes and simplify the welding of these fins to the tubes. With the top-most 3 feet of the each convection tube protruding into the radiant box, the finned portion was basically 19 feet long for a total convection tube length of 22 feet (same as the radiant tubes). The finned portion of the convection tubes was designed to be completely enclosed by a flue gas shell forcing the flue gas to flow between the axial fins. The flue gas shells are further enclosed by insulation. Thus, the convection section in the AFH turned out to be a bank of double-pipe heat exchangers.

Before starting the CFD analyses for the convection tubes, hand calculations were performed using the effectiveness-NTU method to determine the optimum number of fins required around the circumference and the fin height to achieve the target convection heat duty at a reasonable pressure drop (less than or equal to 6 inches WC). These preliminary calculations showed that 32 axial fins would be required around each tube to provide the required duty. A quarter-tube CFD model was developed in order to evaluate the AFH convection box design (Figure 3-38). The quarter-tube model for the convection tube had 8 uniformly-spaced fin channels, or 16 fins, around the circumference, with each fin having a height of 1-1/2” and a thickness of 1/20”. Three 6’4” finned sections along the length of the tube were modeled with small gaps between the sections (something required for manufacturing). As with the radiant tubes, the tube metal and fin thicknesses were included in the model. The tube and the fins were assumed to be constructed from SA-213 T9 steel and 304 stainless steel respectively. The CFD model was run with a flue gas inlet temperature of 1258^{o}F, obtained from the final AFH radiant box model. The CFD simulation for the AFH was performed using FLUENT (v6.2), a general-purpose CFD code. The CFD results showed a net duty of 14.1 MMBtu/hr (which was slightly lower than that from the hand calculations), and a pressure drop of 6.2 inches WC (which was slightly higher than that estimated in the hand calculations). The flue gas exit temperature was calculated to be 786^{o}F (Figure 3-39).

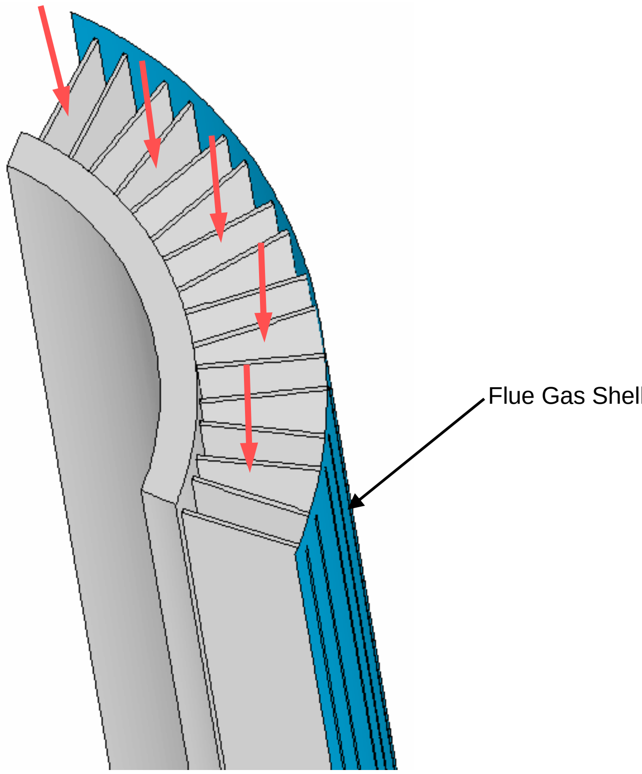


Figure 3-38. CFD Model for the Finned Portion of the AFH Convection Section Tube

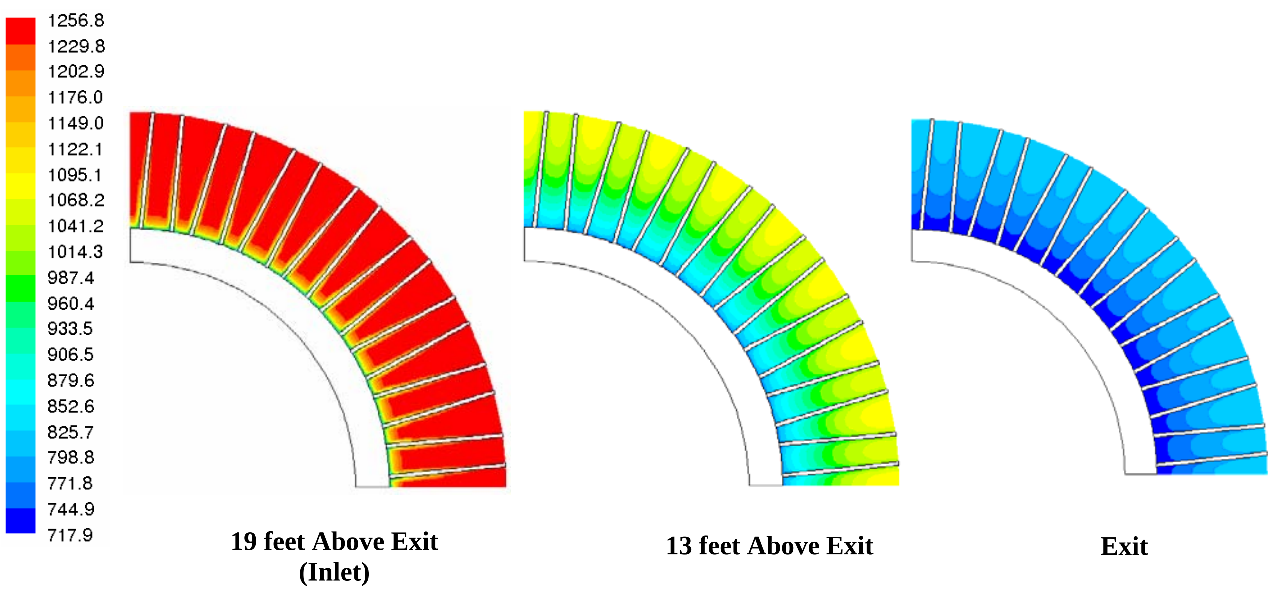


Figure 3-39. Temperature contours (°F) at various elevations in the AFH convection section

The total AFH duty was thus estimated to be 99.2 MMBtu/hr – 80.8 MMBtu/hr from the radiant box, 4.3 MMBtu/hr from the bare portion of the convection tubes, and 14.1 MMBtu/hr from the finned portion of the convection tubes. For the design firing rate of 105 MMBtu/hr, the calculated AFH duty translates to a thermal efficiency 94.5%.

Radiant tubes with back-side studding – a technology considered for the advanced fired heater – was also evaluated as a potential technique for retrofit improvement of overall efficiency of other fired heaters. The tube back-side contributes only about 25% of the total heat transferred to the tube. Extended surfaces may be used on the low-flux tube back-side to reduce the circumferential flux maldistribution and improve the overall heat transfer. Extended surfaces on the tube back-side were expected to produce heat transfer enhancements by increasing the radiative as well as the convective heat transfer. Several fin configurations were considered during the preliminary screening. Although axial fins were considered as a potential enhancement technique during the initial phase of the AFH program, they were not evaluated for other fired heaters due to difficulties in design and fabrication. Studs (or cylindrical spines) were chosen for further analyses, since they can be easily resistance-welded to the radiant tubes in the field. In the fired heater environment, fin efficiencies up to 90% can be obtained with studs. As a part of the AFH program, we analyzed two vertical cylindrical crude oil heaters for the utilization of the studded tube technology – (1) a representative small heater with three John Zink PLNC-15 burners and a firing rate of 20 MMBtu/hr (LHV), and (2) a much larger ExxonMobil heater with sixteen CUB-8 burners and a firing rate of 90 MMBtu/hr (LHV).

A representative small vertical cylindrical crude heater, shown in Figure 3-40, was modeled to evaluate the viability of studded tubes.

CFD models for the heater radiant section were developed with bare tubes and studded tubes for the purpose of performance comparison (Table 3-5). One-sixth of the heater radiant section – with half the John Zink burner – was modeled for the CFD analyses to facilitate the usage of an adequate number of cells around the studs. Two studded-tube configurations were developed – fully-studded tubes with back-side studding along the entire length of the tube (22 ft in the CFD model), and half-studded tubes with back-side studs on the lower half of the tube (11 ft in the CFD model). The 1.25-inch long studs were placed across the rear-most 90^o of the radiant tubes facing the heater refractory (see Figures 3-41, 3-42).

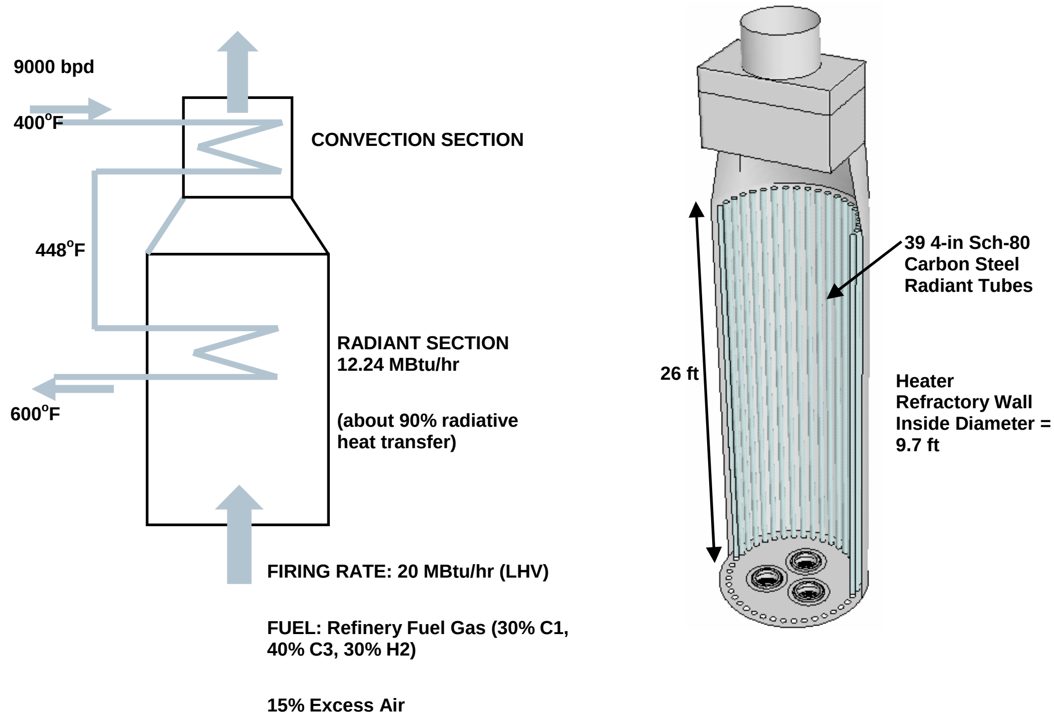


Figure 3-40. Representative vertical cylindrical crude heater evaluated for studded tubes

Table 3-5. Summary of CFD models developed for the representative VC heater

	Bare Tubes	Half-Studded Tubes	Fully-Studded Tubes
Model Size	0.5 Million Cells	1.7 Million Cells	2.8 Million Cells
Computer Resources Used	Two 3.2-GHz CPUs LINUX O/S	Four 3.2-GHz CPUs LINUX O/S	Six 3.2-GHz CPUs LINUX O/S

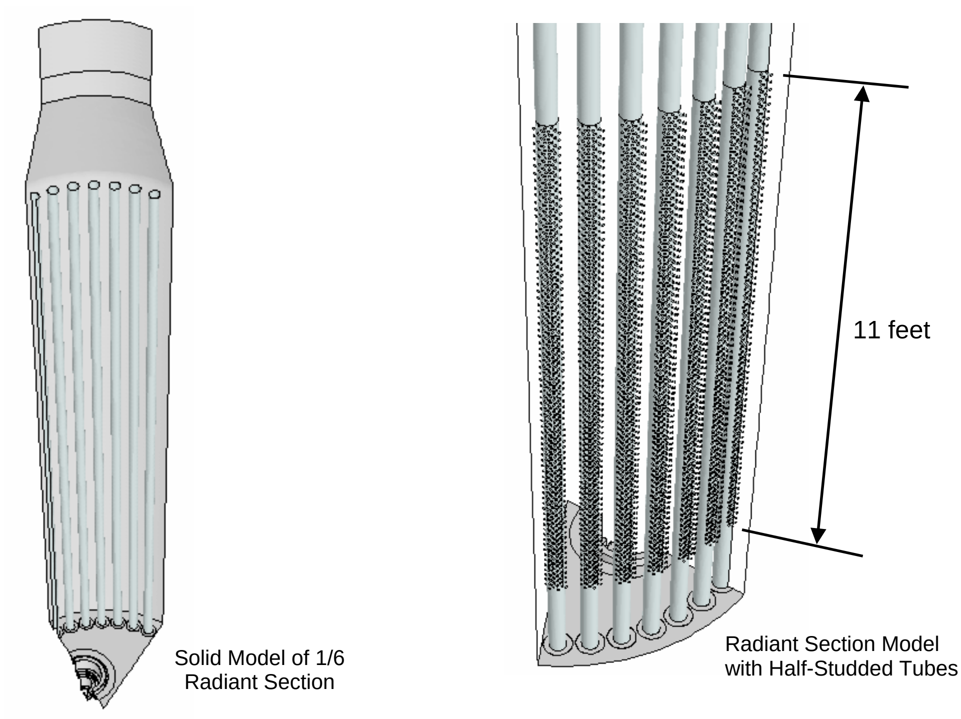


Figure 3-41. CFD models developed for representative VC heater

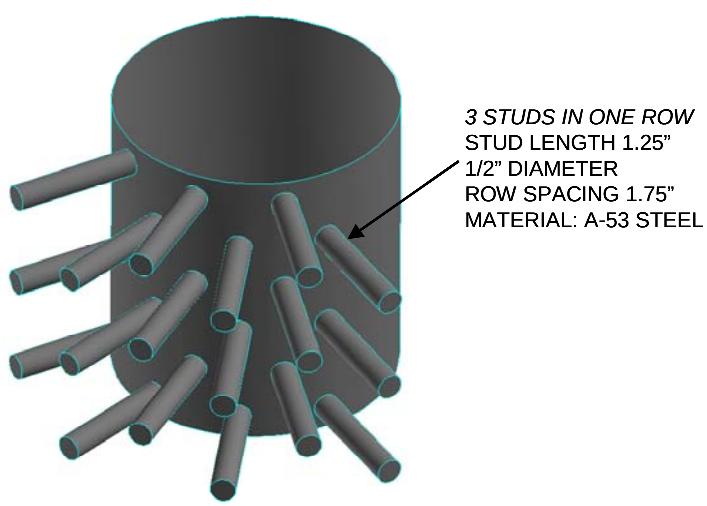


Figure 3-42. Stud Geometry Used in CFD Modeling

After the completion of the CFD runs for all the three models, the results from the studded tube models were compared with those from the bare-tube, baseline model.

Heat transfer in both the radiant section models with studded tubes showed improvements. However, the CFD results also exhibited differences in the heater flow field for the studded-tube cases; the flow trajectories in the studded-tube cases showed a distinct turn toward the tubes (see Figure 3-43). The changed flow field had a positive impact on the flame-side heat transfer in half-studded case. This higher flame-side heat transfer combined with the enhanced heat transfer on the tube back-side resulted in the half-studded case showing the highest radiant section efficiency.

For the fully-studded case, however, the changed heater flow structure affected the flame-side heat transfer negatively – in fact, lower than that in the baseline case. This is clearly seen in Figure 3-44, which shows a rapid decline in the flame-side heat flux at heater elevations greater than 16 feet for the fully-studded case. Although the contribution of the studs toward increasing the heat transfer on tube back-side was the highest in the fully-studded case, the lower flame-side heat transfer led to a lower net heat absorption in the radiant section when compared to the half-studded case (see Figure 3-45). The CFD results for all the three cases are summarized in Table 3-6.

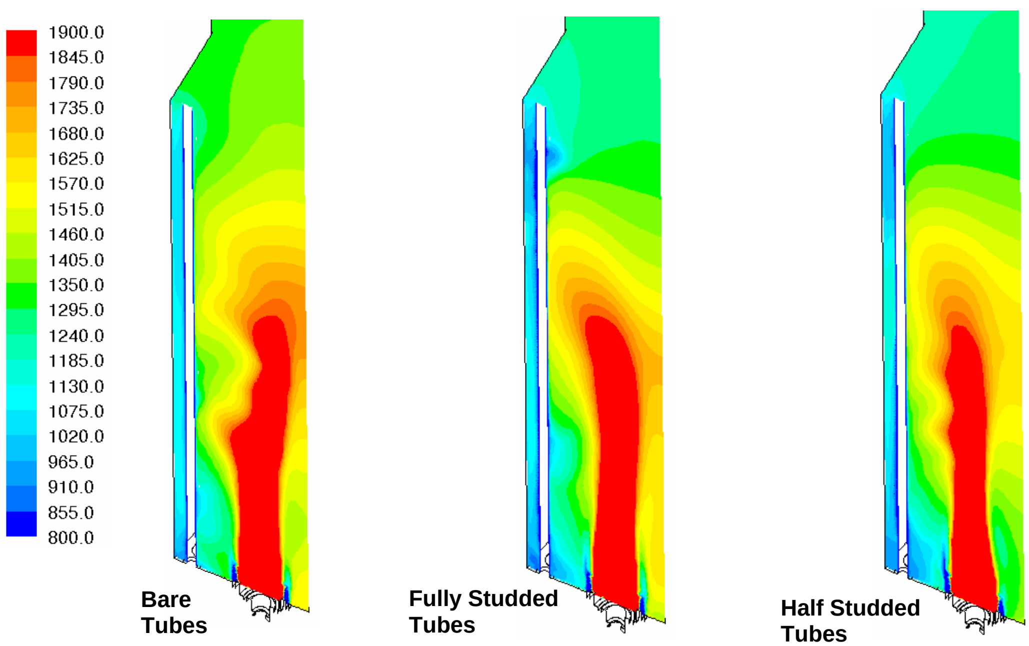


Figure 3-43. Contours of temperature observed during the CFD simulations

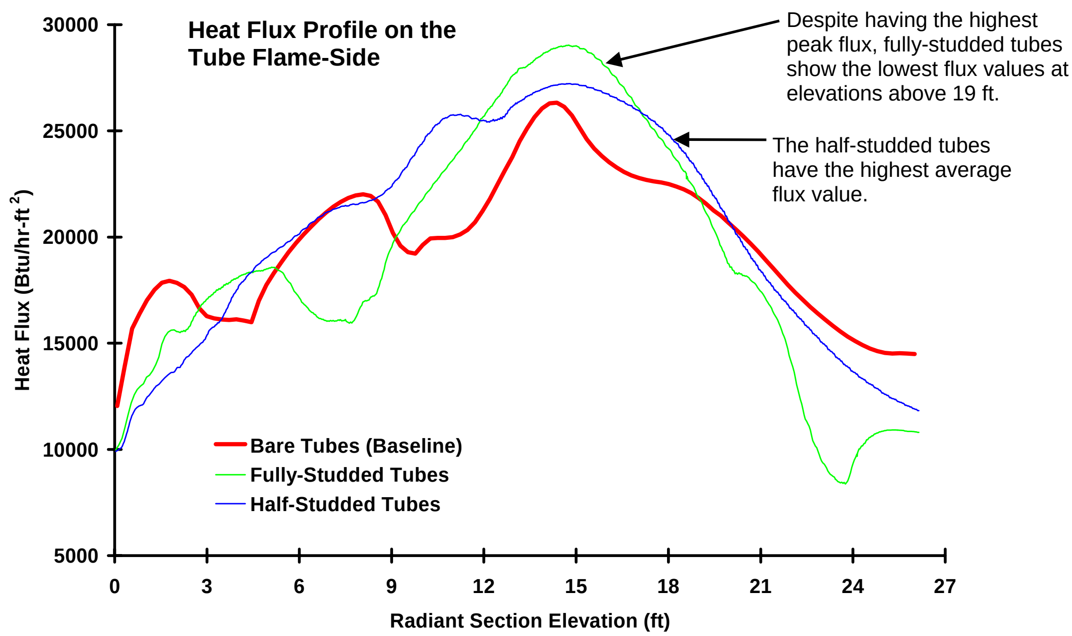


Figure 3-44. Heat flux distributions on a representative radiant tube

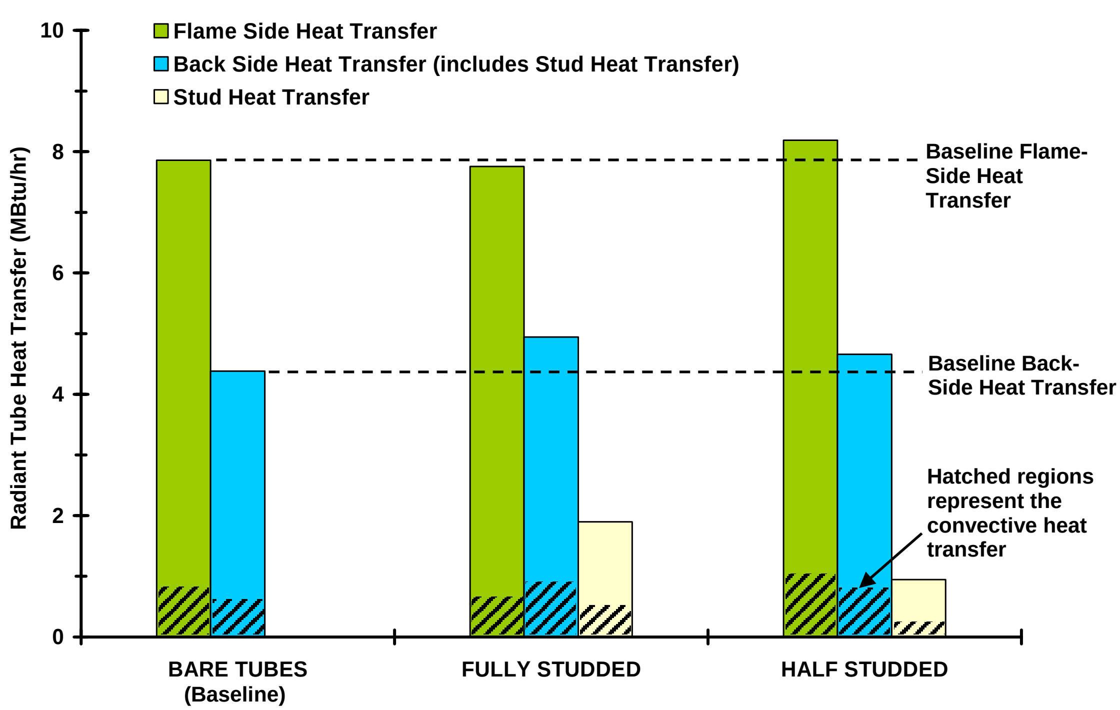


Figure 3-45. Summary of heat flux distributions on a representative radiant tube

Table 3-6. Summary of CFD results for the representative VC heater

	Bare Tubes	Half-Studded Tubes	Fully-Studded Tubes
Heat Transfer Through Uncovered Tube Surface	12.2 MMBtu/hr	11.9 MMBtu/hr	10.8 MMBtu/hr
Heat Transfer Through Studs	N/A	1.0 MBtu/hr	1.9 MBtu/hr
Total Heat Transfer in Radiant Section	12.2 MMBtu/hr	12.9 MMBtu/hr	12.7 MMBtu/hr
Percentage Heat Transfer Through Studs	N/A	7.8%	15%
Radiant Section Efficiency	61.2%	64.3%	63.5%
% Increase in Radiant Section Heat Transfer	N/A	5%	3.8%

A second heater – a vertical cylindrical pipestill heater at an ExxonMobil refinery – was also evaluated for the utilization of studded tubes. This heater was much larger compared to the representative heater considered earlier. The second heater operates in the forced-draft mode with preheated combustion air at 570^{o}F and uses a wide range of fuel gases from CO-rich, low-Btu gases to C3/hydrogen-rich refinery fuel gases.

The modeling approach remained the same; however, this time one-sixteenth of the heater was modeled with one complete CUB-8 burner (Figure 3-46). Four CFD models for the second heater were developed – a baseline CFD model with bare tubes, a CFD model with back-side studding on the lower 16 feet of the radiant tubes (half-studded), a CFD model with back-side studding on the lower 25 feet of the radiant tubes (three-fourth studded), and a CFD model with back-side studding along the entire tube length (fully studded). As before, the 2.5-inch long studs were placed across the rear-most 90^{o} of the radiant tubes facing the heater refractory (Figure 3-47).

The CFD results for the fully-studded tubes and the three-fourth studded tubes were observed to be substantially different from those for the baseline case with bare tubes. This is clearly seen in Figure 3-48. The flow fields for the fully- and half-studded tubes show the formation of a rather large recirculation zone adjacent to the radiant tubes. This flue gas recirculation is likely due to a relatively colder downdraft caused by the studded tubes. The studs on the tube back-side extract more heat from flue gases creating a colder, and hence denser, region behind the tubes. These gases descend rapidly between the tubes and the wall pushing the flame away from tubes and resulting in the formation of a large recirculation zone that extends almost along the entire length of the tube. Consequently, the benefit of the additional heat absorbed by the tubes is lost

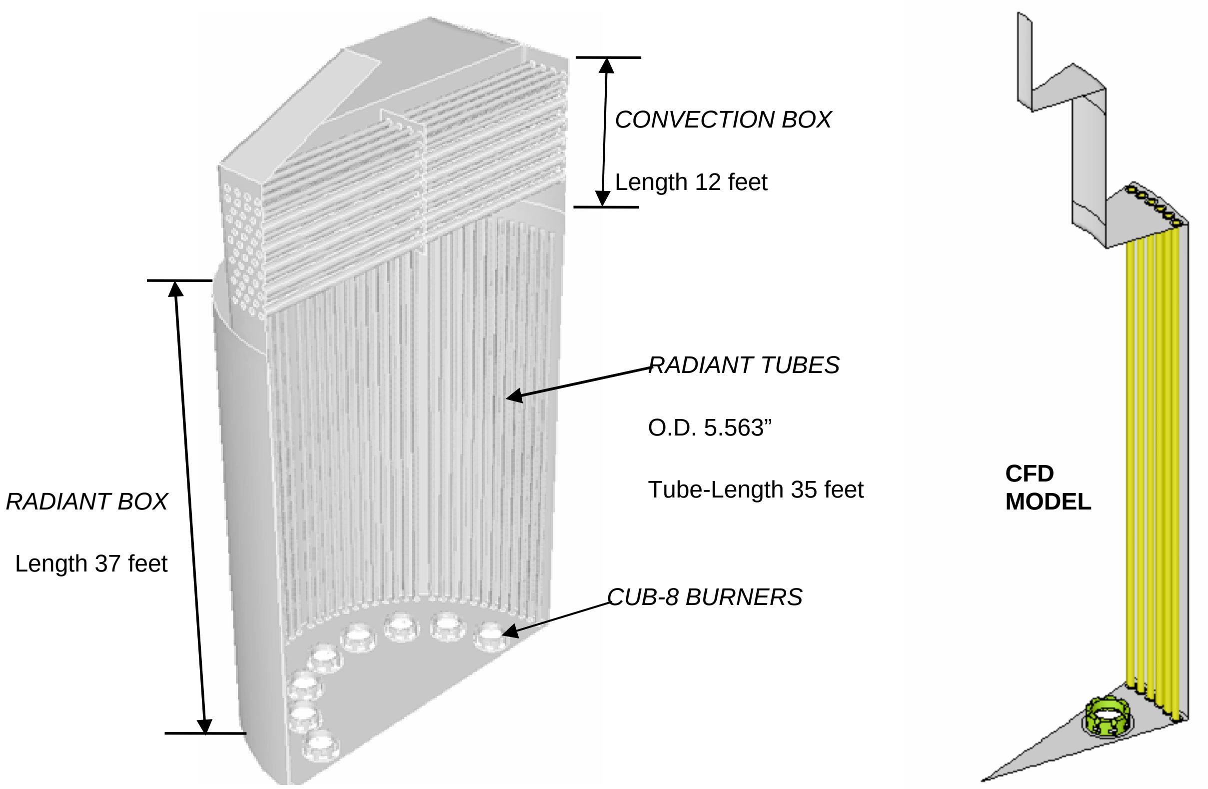


Figure 3-46. Schematic of the ExxonMobil vertical cylindrical heater evaluated for studded tubes, and the CFD model

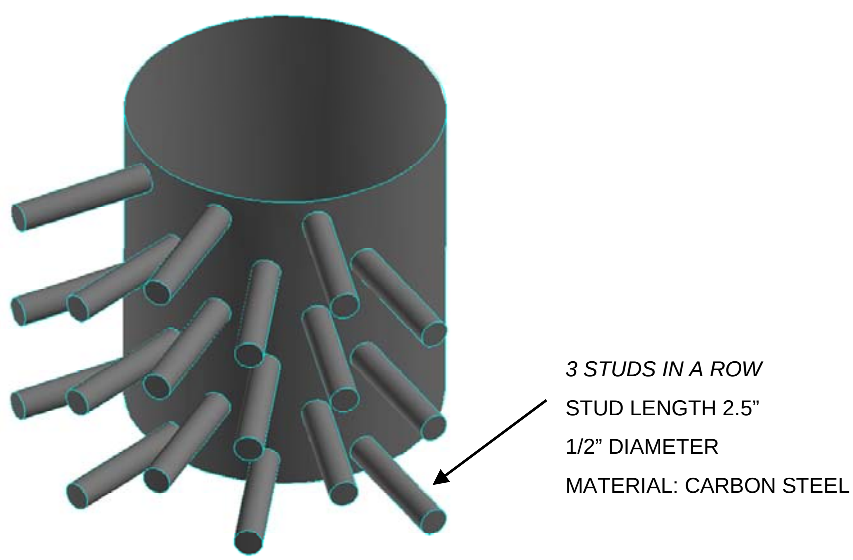


Figure 3-47. Studs used in the heater modeling

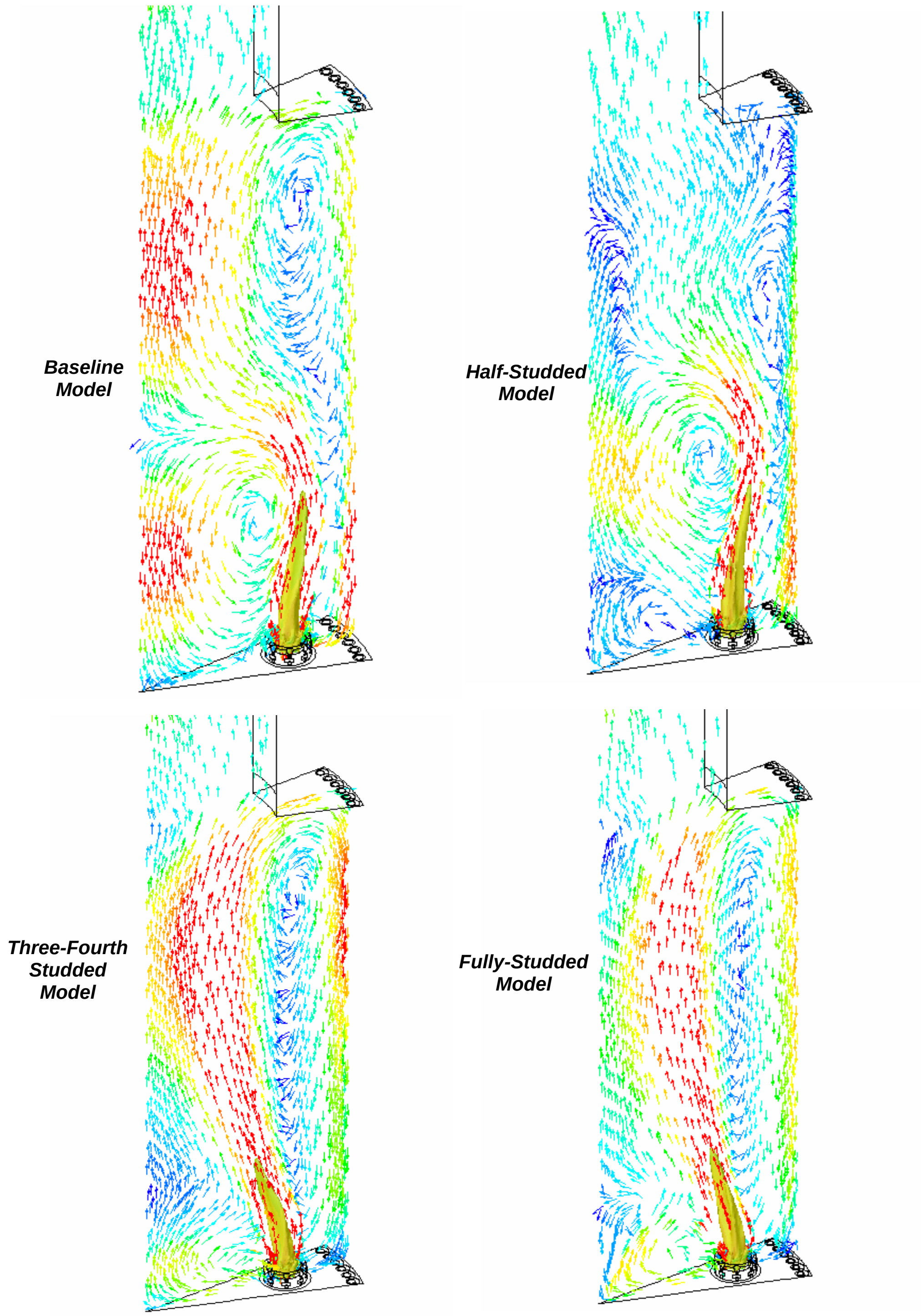


Figure 3-48. Flow structures observed during the CFD simulation of the ExxonMobil vertical cylindrical heater with studded tubes

as a result of the poor heat transfer on the tube flame-side that is significantly lower than the baseline. As seen in Table 3-7, the three-fourth studded and fully-studded cases have lower radiant section efficiencies and much higher bridgewall temperatures.

Table 3-7.   Summary of CFD results for the vertical cylindrical heater

	Bare Tubes (Baseline)	Half-Studded Tubes	3/4-Studded Tubes	Fully-Studded Tubes
Heat Transfer Through Uncovered Tube Surface	52.8 MMBtu/hr	52.6 MMBtu/hr	45.6 MMBtu/hr	44.5 MMBtu/hr
Heat Transfer Through Studs	—	4.2 MMBtu/hr	2.7 MMBtu/hr	3.7 MMBtu/hr
Net Heat Transfer in Radiant Section	52.8 MMBtu/hr	56.8 MMBtu/hr	48.3 MMBtu/hr	48.2 MMBtu/hr
Radiant Section Efficiency	58.7%	63.1%	53.7%	53.5%
% Change in Radiant Section Heat Transfer	—	7.6%	-8.5%	-8.8%
Bridgewall Temperature	1610^{o}F	1480^{o}F	1780^{o}F	1785^{o}F
Peak Heat Flux	40200 Btu/hr-ft^2	46000 Btu/hr-ft^2	26900 Btu/hr-ft^2	25200 Btu/hr-ft^2

Only the half-studded case showed a flow field similar to the baseline bare-tube case. The similar flow field in the half-studded model brought about an almost equivalent flame-side heat transfer, which, combined with extra heat absorbed by the studs on the tube back-side, resulted in a net increase in the total heat absorbed. However, the concerns with this case were the considerably lower bridgewall temperatures (that could diminish the convection section performance and therefore the overall heater efficiency), and the relatively higher peak heat flux value.

As a result of these parametric evaluations of enhanced heat transfer, it was concluded that studded tubes, and more specifically half-studded tubes, can be used to improve the efficiency of the radiant section in fired heaters. The CFD results for the two vertical cylindrical heaters evaluated during this program showed a reasonable increase in the radiant section efficiency with half-studded tubes. However, these gains may be offset by reduced convection section efficiency due to the lower bridgewall temperatures. Generally speaking, while studded tubes do enhance heat transfer to the tube back-side, our modeling results also showed that the flow pattern in a heater is likely to be affected by the studded tubes. The modified flow and temperature fields can have an impact on the flame-tube interaction, which in turn, can potentially reduce the tube flame-side heat transfer. Moreover, the level of impact on the heater flow field is going to be dependent on the specific heater geometry under consideration, its operating conditions, the burners used in the heater, and the configuration of the studded tubes. For conventional new heaters, studded tubes can be used as a means to achieve a more compact radiant section design. When retrofitting existing heaters with studded radiant tubes, however, a comprehensive evaluation of the entire system (including the convection section, air preheaters and downstream heat recovery systems) needs to be performed before making the final decision to install the studded tubes.

3.3 Tube Metal Temperature Monitoring

The initial intent of the TMT monitoring was to integrate an on-line control system to the burner and combustion controls for the advanced fired heater. After preliminary evaluation of the scope, operational complexity and cost of this approach, a reduced goal of providing real-time non-intrusive measurement of multiple tube locations was selected. This information will allow operators to adjust combustion conditions to control and mitigate the effects of poor operation that often lead to poor heat flux distribution, flame impingement and coking. The need for better diagnostic information is especially acute when operating at ultra low NO_x conditions, when the flame envelope is frequently operated in an off-design mode to reduce emissions. Currently available temperature monitoring technology includes the installation of many thermocouples in the furnace, or periodic monitoring with handheld thermal imaging cameras to diagnose heater conditions. The objective of the current work was to improve on the reliability and response time of this technology.

The effort in the DOE project was merged to a joint development effort between Honeywell Process Solutions and ExxonMobil Research and Engineering Company to develop a fired heater monitoring system. This work was centered on multiple, permanent infrared imaging cameras to allow more comprehensive characterization of the furnace thermal environment than is possible with thermocouples. ExxonMobil and Honeywell identified low cost options for imaging cameras using Honeywell infrared detector technology. One cost savings feature of the imaging camera is operation in a shapshot mode to give discrete temperature readings, rather than a continuous image mode. The concept is that the imaging camera would be mounted on a movable mechanical system and inserted into the furnace at prescribed intervals. IR snapshots at numerous tube locations would be processed into temperature readings and transmitted to the control room. At a minimum, the temperature data would be used for TMT alarm generation and archival trending to detect fouling or coking patterns.

The work involved the design of a mechanical translation system to insert a thermal imaging device into a fired heater and to collect temperature information for components inside the fired heater. The system design included a positioning mechanism to activate and control the motion of the enclosure and imaging components within the enclosure, a infrared camera to collect thermal images, a telemetry system to transmit signals via a wireless modem to a remote location, and a data acquisition system to archive and process the data. Once system design was completed, the system was installed in a gulf coast refinery and tested over several months. The intellectual property resulting from this effort is owned by Honeywell, and an application has been filed with the U. S. Patent Office.

3.3.1 System Design:

The system design had 3 main functional areas: a) Mechanical motion actuation and control, b) Thermal imaging system data collection and transmission and c) Data archiving and visualization.

The initial design effort focused on assuring the survivability of an infrared camera inserted into the radiant section of a fired heater. A stainless steel air purged enclosure was constructed to house the infrared camera system. An infrared camera was installed inside the enclosure and attached to a tube several feet in length for manual insertion into an operating furnace. It was found that by limiting the high temperature exposure time only to the time required to collect an image, the maximum operating temperature rating for the electronics was not exceeded. Using this criteria, the control system for the insertion of the imaging system into the furnace was developed.

The prototype system was designed and constructed with provision for 3 independent motions. The first motion was the opening of an access port for the insertion of the infrared imaging camera. Linear actuators were used to raise an insulated door mounted to a frame outside the furnace. Second, once the door was raised, the field of view was selected by rotating the thermal imaging camera enclosure. The enclosure housing the camera was mounted at an off axis angle that allows views of burners (on the furnace floor), furnace tubes on the lower left, upper left, directly above, upper right, or, lower right. The angle positioning was accomplished with a stepper motor. Finally, once the viewing angle was set, the system was inserted in the furnace with an additional linear actuator. Once the system was completely inside, the image data commenced. After the prescribed data acquisition, the system was then withdrawn from the fire box and the port closed. With this design, the enclosure required about 30 minutes to cool to ambient temperatures once withdrawn from the furnace. Unless active cooling is provided, this could limit the frequency of data collection. All the linear actuators were pneumatic and controlled with a programmable logic controller. Figure 3-49 shows the system installed at a refinery heater.

The data collected during imaging camera insertion was transmitted via a wireless modem to a location approximately 0.5 miles from the furnace. Local atmospheric conditions required a change in the wireless design in the field, which employed some software modification and use of a directional antenna.

Figure 3-49. Positioning System for Infrared Imaging Camera

The data were archived in a desktop computer located at the remote location with a writable compact disk for manual data transfer. The thermal imaging data was transmitted as an array of values, which was converted to a temperature reading. The arrays were then used to construct a thermal image, with software provided by the IR camera manufacturer. No local area network connections were put in place for these field trials.

3.3.2 Field Test Results:

The mechanical positioning system was operated for periods of several months. The initial system had under sized actuators for both the door opening mechanism and the camera insertion. The undersized actuators caused the control system to time out and abort due to slow translation. Once the actuators were upgraded to provide an increase in force, no further problems were experienced either in opening the access door or inserting the imaging system into the fire box.

The stepper motor used to select the furnace view was over sized. This led to some instances of excessive temperatures in the enclosure housing the motor controller, especially in the summer months. Through a programming change in the controller this was also resolved. Overall the mechanical system proved to be reliable, once the initial design deficiencies were corrected.

Due to atmospheric conditions at the test site, communications failures were experienced early in the testing. Two steps were taken to correct the problem. First a software modification was made to allow several attempts to re-establish communications. This improved the availability of wireless transmission, but did not completely eliminate the problem. The second step taken, the installation of a directional antenna, resolved the problem. Once these two changes were made no further communications failures were experienced.

The infrared imaging system did not work well. The imaging camera employed was a thermo-electric technology. In the furnace application, this imaging technique revealed two limitations. First, the imaging camera employs a semiconductor array that is highly reflecting. Internal reflections within the enclosure caused a ghosting of the image. To resolve this issue a set of internal baffles were installed, which successfully removed the image ghosting.

The second issue proved to be a fundamental limitation of this imaging technique. The thermoelectric technology is affected by thermal gradients. Even though the maximum operating temperature for the system was never exceeded, the thermal gradients created inside the enclosure during insertion into the fire box, led to image banding. Banding is defined as a bias in the infrared image that appears as bands in a group of rows (pixels). Post processing of the data after collection was attempted unsuccessfully. This was mainly due to the fact that the bias was variable and unpredictable.

It was concluded that the existing system design (mechanical system, control, and data acquisition) worked well and is suitable to the TMT monitoring purpose, but that a different infrared imaging technology such as a micro-bolometer based camera is needed for the process heater furnace environment.

Additional work would be required to develop the machine user interface and develop control applications that use the thermal data as the basis.

In addition, it was found that it is quite feasible to monitor burner fouling even with the thermoelectric technology.

