**A Novel High-Heat Transfer Low-NO_x
Natural Gas Combustion System**

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Phase I
Final Report

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November 1997

Prepared for
The U. S. Department of Energy
Under Award No. DE-FC02-95CE41185

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Executive Summary

This report is submitted to the United States Department of Energy in partial fulfillment of the contractual requirements for Phase I of the project titled, 'A Novel High-Heat Transfer Low-NO_x Natural Gas Combustion System', under co-operative agreement number DE-FC02-95CE41185. The full project is divided into three Phases with go-no go decision points after Phases I and II. The 15-month Phase I of the project, titled Research and Development Definition, focused on acquiring the market needs, modeling, design, and test plan information needed to proceed with Phase II. Phase I was completed on time and within budget, and Phase I was highly successful. All goals and objectives for Phase I were achieved, and the project team is excited about the commercial potential for this innovative new technology.

The project team assembles all elements necessary to develop and demonstrate this novel, innovative technology and to accelerate its commercialization. The Institute of Gas Technology (IGT) is a premier industry-supported energy systems and environmental technology development organization. Combustion Tec, Inc. (CTI) is a major supplier of advanced natural gas combustion systems to the glass industry. Owens Corning (OC) is a leading glass industry manufacturer/end-user with a well-established reputation for the introduction of innovative technology. IGT and Combustion Tec have an established track record of successful teaming arrangements over the past fourteen years that has rapidly commercialized innovative technologies in the glass industry. Owens Corning participation ensures development is focused on end-user needs and commercialization barriers are minimized. The strong industrial participation in this project team and the active participation of industrial team members in all Phases and Tasks of the project demonstrates the commitment of industry to develop and commercialize this innovative energy-saving, low emission combustion system.

The key component of the system is an innovative burner technology conceived by IGT. This novel burner capitalizes on IGT's and Combustion Tec's extensive expertise in this area.. This technology, which combines high temperature natural gas preheating with soot formation and subsequent soot burnout in the flame, increases the system's energy efficiency and furnace throughput, while minimizing the furnace air emissions, all without external parasitic systems. The burner will be a highly cost-effective retrofit to existing high temperature furnaces fired with natural gas using oxygen, oxygen-enriched air, or air.

The framework for the project has been set in Phase I, the R&D definition phase, which is now completed. The results of Phase I are presented in this Phase I Final Report. Work has included identifying industry's needs and constraints, modeling the high luminosity burner system, designing the prototype burner for initial laboratory-scale testing, defining the test plan for Phase II, adapting the IGT burner technology to meet the industry's needs and constraints, and outlining the Industrial Adoption Plan. Owens Corning, Combustion Tec and IGT have all participated in this phase of the project.

The modeling results produced by the University of Illinois at Chicago (UIC) and Purdue University were very positive. These teams modeled the burner system as consisting of a Fuel Preheat Zone and A Flame Zone. They determined that the direct heating in the Fuel Preheat Zone and the subsequent soot precursor formation increases the carbon conversion to soot in an oxy-gas flame from 0.1 percent of the feed carbon to 2 percent of the carbon. Flame radiation heat transfer calculations show this leads to an increase in heat transfer to the load of 25 percent. Only 2 to 9 percent of the feed natural gas must be burned in the Fuel Preheat Zone's

Precombustor to achieve this effect, and the residence time needed to generate the soot precursors is as short as 10^{-4} seconds. A Topical Modeling report has been prepared and submitted to DOE. The CTI burner design team developed a prototype burner design for laboratory testing which covers the temperature and residence time ranges specified by the modeling results. This design can be readily fabricated by standard practice, and the burner is not expected to be outside the physical size range of existing oxy-gas burners.

The experimental portion of the program will begin in Phase II, the Research and Development phase, by engineering and testing at laboratory and pilot scale, the IGT burner technology. These proof-of-concept tests will assure that the burner technology will meet industry's technical and financial needs and constraints. The testing will take place in IGT's Applied Combustion Research Facility (ACRF). Combustion Tec will construct the test burners, Combustion Tec and IGT will analyze the test results, and Owens Corning will review all results to ensure that the test results meet the industry's needs and constraints.

The Phase III full scale demonstration at an Owens Corning facility, will confirm the suitability of the burner technology and provide the information necessary to complete the Industrial Adoption Plan. All team members will participate in Phase III of the project.

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1.0 Introduction

The two largest problems facing high temperature combustion processes are the consumption of large quantities of fuel and the production of high levels of pollutants. The production of NO_x is of particular concern in high temperature processes because NO_x yield increases as combustion temperature increases. Both problems can be addressed for natural gas combustion by switching from air-gas to either oxygen enriched air-gas or oxygen-gas firing. Combustion with oxygen offers advantages of reduced energy costs, lower NO_x and particulate emissions, and decreased capital and maintenance costs by eliminating the need for flue gas cleaning and heat recovery. Oxy-gas flames burn at a higher temperature, allowing energy savings through more efficient radiant heat transfer and higher energy availability. Low NO_x levels result from the presence of a reduced amount of nitrogen in the flame.

While oxy-gas firing for industrial processes such as glass melting is attractive and has been adopted commercially to some extent, optimization of the process promises large rewards in both energy savings and NO_x reduction. The dominant mode of heat transfer to the load in a process such as glass making is by radiation. Radiative heat transfer is dependent upon both the temperature and emissivity of the heat source. At the high combustion temperatures of an oxy-gas or a high temperature air-gas flame, the combustion products are not luminous (low emissivity) which leads to low radiation, poor heat transfer to the batch, a high exit gas temperature, low process efficiency and production rate, high fuel consumption, and poor furnace efficiency. Process and energy efficiency increase for high temperature furnaces. Increasing flame luminosity by seeding the flame with soot particles has been shown to provide significant benefits to oxy-gas combustion.^{1,2} Although huge potential gains can be made, commercial application of this technology has not occurred due to the complexity and cost of co-burning a source of soot particles with natural gas.

The high-heat transfer low- NO_x natural gas burner system for oxy-gas furnaces produces an improvement in energy efficiency of at least 20% and a significant decrease in NO_x formed per ton of product. In this innovative burner, improvements arise from increased luminosity produced by forming and then consuming soot in the flame. Benefits include:

- Higher heat transfer to the load. Cracking of natural gas generates soot particles which increase luminosity and provide higher heat transfer as well as higher process and energy efficiency.
- Lower flame temperature and exit temperature. Greater heat transfer from a more luminous flame generates a radiative cooling effect.
- Lower NO_x yield. The lower flame temperature and the lower consumption of natural gas lead to less NO_x production

The burner system can be retrofit to existing air-gas fired furnaces minimizing cost and maximizing energy savings, production rate, and NO_x reduction.

The burner being developed in this program, referred to as the IGT high-heat transfer low- NO_x natural gas burner, has direct application in glass melting furnaces and other industrial furnaces. The burner combines a preheating zone and two combustion zones. A conceptual drawing of the burner is presented in Figure 1. The high-heat transfer low- NO_x natural gas burner has the following zones:

- Preheating zone - includes partial combustion and direct preheating of natural gas
- First combustion zone - soot formation, fuel-rich partial combustion
- Second combustion zone - soot burning, high luminosity combustion.

Conventional industrial oxygen-natural gas combustion furnaces offer little, if any, energy efficiency improvement over a glass tank with a high air preheat temperature (2300°F) from a good regenerator.³ A substantial amount of energy is needed to separate air and produce industrial oxygen. Significant reductions in NO_x production can be realized, but only at the expense of using pure oxygen instead of industrial oxygen with several percent nitrogen. The natural gas must also be free of nitrogen to achieve low NO_x levels. Natural gas often contains large amounts of nitrogen (3 - 10%). To derive the full benefits of oxygen-gas combustion in

Figure 1. HIGH-HEAT TRANSFER LOW- NO_x NATURAL GAS BURNER CONCEPT

commercial processes, cost effective heat recovery technologies and a means to use industrial oxygen are needed. Energy savings realized by preheating natural gas alone with a recuperator are not cost effective. However, using preheated (and possibly partially reformed) natural gas to generate soot in a first stage of combustion and a highly luminous flame in a second stage of combustion provides additional benefits of lower NO_x emissions, higher heat transfer to the glass, increased productivity, reduced furnace size, and substantially reduced furnace heat losses through the wall.

This technology offers immediate, realistic benefits to industry. These include:

- Applicable to a wide range of furnaces in the materials processing industry, including glass melting, metal melting and treatment, and cement production
- Enables industry to meet emission regulations cost effectively, while also increasing production rate and system thermal efficiency
- Can be retrofit to existing furnaces without major furnace modifications
- Allows utilization of more cost efficient industrial oxygen, instead of pure oxygen, without generating high levels of NO_x .

Particles in a flame are known to cause a flame to burn with greater luminosity than if the particles are not present. Conductive and convective heat transfer raises the particles to a high temperature after which they radiate as nearly perfect black bodies. Soot particles are present in coal and oil combustion but not in less luminous natural gas combustion.

In the high-heat transfer low- NO_x natural gas burner, soot particles are created and burned in the flame. This increases the flame luminosity, and thus the heat transfer to the load, while producing lower NO_x and other air emissions such as CO, CO_2 , and unburned total hydrocarbons. The higher heat transfer will lower the flame temperature and decrease the NO_x emissions.

The High-Heat Transfer Low- NO_x Natural Gas burner improves upon IGT's proven oxygen-enriched two-stage low NO_x burner and the Combustion/IGT oxy-natural gas cracker developed and proven by IGT and Combustion Tec to increase the flame luminosity and thus the heat transfer from the flame to the load, thereby increasing the furnace production rate and decreasing NO_x emissions. The involvement of Combustion Tec, Inc., the primary supplier of combustion technology to the U.S. glass industry, in this project assures that the developed technology will be commercialized in the glass and other industries. Owens Corning, the third team member, is a major international producer of fiberglass. Their participation in this project confirms the strong industry interest in the development of this technology.

Combustion Tec and IGT previously invented a separate oxy-natural gas cracker which produces soot particles. The particles from the external cracker are transported to a low NO_x burner where they are burned along with the natural gas. Flame luminosity was increased but the system cost was also increased as a result of the cost of the cracker, and system performance was not optimized. IGT has run experiments which show that the heat transfer from the flame is increased when soot particles are present. This can only occur if the flame luminosity is increased by the presence of the particles.

Significant work by IGT and others supports the concept of this innovative IGT burner. Recent work by Factory Mutual indicates methane needs to be preheated to 1000°C (1830°F) and above to create significant amounts of soot in the flame.⁴ Figure 2 shows high methane preheat temperature significantly increases flame radiation. Older work of Wolanin indicates the optimum methane preheating temperature is in the range 1150° to 1180°C (2100° to 2160°F).⁵ Wolanin shows the heat transfer to the load is increased by 63% when 26.5% of the natural gas is decomposed and the gas is preheated to 1190°C (2174°F). To have a practical high luminosity burner with soot generated in the flame, these results indicate the natural gas must be preheated to between 1000° and 1180°C (1830° and 2160°F) without cracking and then mixed with oxygen. The desired soot should be formed during the oxygen-hot natural gas combustion, and not during the heating of the natural gas. The natural gas needs to be heated very rapidly to form soot precursors. Significant amounts of carbon particles can nucleate and grow into soot when natural gas containing soot precursors is mixed with the oxygen.

Figure 2. RADIANT FRACTION MEASUREMENTS AS A FUNCTION OF THE FUEL PREHEATING TEMPERATURE AT VARIOUS OXYGEN PREMIXING FRACTIONS

Numerous models of methane combustion exist in the literature. These models range from complex to very simple kinetic models. The complex models include detailed kinetics of all major and minor flame species⁸ and typically involve reaction schemes with 75 to 150 reactions and 30 to 50 reaction species. Simple models have typically had 1 to 20 reactions⁹⁻¹⁰ and involve <10 of the major flame species (i.e., CO₂, H₂O, H₂, CO, C₂H₂, H, and OH). Both types of models can be employed for predicting major species concentration, temperature profiles, and laminar burning velocities.⁶⁻¹¹ In simplified models, species whose concentrations are not calculated by the kinetics scheme are calculated by assuming thermodynamic equilibrium exists between calculated species and those not calculated. Our efforts focused on selecting and modifying one of the existing simple models such that the model could accurately predict not only the major flame gas and temperature profiles, but also the concentration profile of the soot nucleus and growth species. The selection of a soot nucleus and growth species and their kinetic

modeling along with modeling of the radiation heat transfer from the flame constitutes a major contribution of this Phase of the program. A separate Topical Report on modeling has been preparing and includes detailed modeling of the Fuel Preheat and Flame Zones of the high luminosity burner performed by the University of Illinois at Chicago and Purdue University.

High temperature industrial combustion processes often have relatively low energy efficiency and generate high levels of NO_x emissions. These problems are particularly acute in the glass industry which consumes 200 bcf/y of natural gas and is reported to be the fourth largest industrial energy consumer in the U.S. Work in this program addresses the glass industry as a first priority because of the large positive impact the technology would have on the industry, improvement in energy efficiency, and decrease in NO_x production.

Regenerative glass furnaces use extremely high combustion air temperatures (2000° to 2400°F) to improve production rate, product quality, and furnace thermal efficiency. Furnace and flame temperatures - and consequently NO_x generation - are high. NO_x emissions of over 3000 vppm (12 to 15 lb/ton of glass produced) are not uncommon from natural gas-fired glass furnaces.^{12,13} There are no current national regulations on NO_x emissions from glass tanks in the U.S., but this could change in light of the 1990 Clean Air Act. These emissions are restricted in some areas - the most stringent being Southern California. The South Coast Air Quality Management glass District currently restricts NO_x emissions from glass furnaces to 4.0 lb/ton of produced.¹⁴ Even stricter regulations are now being implemented for this region, requiring NO_x reduction to below 0.8 lb/ton in the year 2003. The current 4.0 lb/ton regulations can be approached using relatively simple combustion modification techniques developed by IGT and Combustion Tec (with funding support from GRI and SoCalGas).^{13,15} Augmenting the electric boost and increasing the percent cullet in the feed can also decrease NO_x emissions by decreasing the natural gas combusted.

Other technologies have been developed to decrease NO_x emissions from glass furnaces.^{16,17} The various methods are listed in Table 1. While some of these methods can remove a large fraction of the NO_x formed, all of them except Oxygen-Enriched Air Staging and oxygen-fuel firing have no affect or decrease the efficiency of the glass making process. Fuel oil does produce some reduction in NO_x formation but at the expense of higher SO_x and particulate emissions, higher fuel costs, higher vanadium and sulfur concentrations, and decreased furnace life because of higher furnace crown temperatures.

The glass industry would benefit greatly from the adoption of the High Luminosity Burner technology since it would be an easily installed process modification which would increase process and energy efficiency by more than 20% while significantly decreasing NO_x emission levels at a relatively low cost. The high-heat transfer low-NO_x natural gas burner can achieve both goals for oxygen-, enriched air-, and air-fired furnaces. The greatest energy efficiency gains will be seen for oxy-gas fired furnaces because current technologies are still far from fully exploiting the potential benefits of oxy-gas combustion in terms of flame luminosity, heat transfer increase, NO_x reduction, and costs. An important feature of the high-heat transfer burner is that it can be used in conjunction with most NO_x reduction techniques including combustion modifications and Oxygen-Enriched Air Staging.

Table 1. COMPARISON OF NO_x REDUCTION TECHNOLOGIES

<u>NO_x Reduction Technology</u>	<u>NO_x Reduction, %</u>	<u>Relative Cost</u>	<u>Efficiency</u>
Cullet Preheating	5	Moderate	Small Increase
Electric Boosting	30	High	Decrease
Selective Non-Catalytic Reduction	30	Moderate	No Change
Fuel Oil Switching	30	Moderate	Small Increase
Fuel Staging	35	Moderate	No Change
Oxygen-Enriched Air Staging	60	None to Low	Small Increase
Reburning	60	Moderate	Decrease
Selective Catalytic Reduction	75	High	No Change
Oxygen-Fuel Firing	80	Moderate	No Change
High-heat Transfer Low-NO _x Burner for Oxy-Fuel Firing	90+	Moderate	Large Increase (> 20%)

2.0 Objectives

The objectives of this program are to research, develop, test, and commercialize a novel high-heat transfer low-NO_x natural gas combustion system for oxygen-, oxygen-enriched air, and air-fired furnaces. This technology will improve the process efficiency (productivity and product quality) and the energy efficiency of high-temperature industrial furnaces by at least 20%. IGT's high-heat transfer burner has applications in high-temperature air, oxygen-enriched air, and oxygen furnaces used in the glass, metals, cement, and other industries. Development work in this program is focused on using this burner to improve the energy efficiency and productivity of glass melting furnaces which are major industrial energy consumers. The following specific project objectives are defined to provide a structured means of achieving the overall project objectives..

- Identify topics to be covered, problems requiring attention, equipment to be used in the program, and test plans to be followed in Phase II and Phase III
- Use existing codes to develop models of gas combustion and soot nucleation and growth as well as a thermodynamic and parametric description of furnace heat transfer issues
- Conduct a parametric study to confirm the increase in process and energy efficiency
- Design and fabricate a high-heat transfer low-NO_x natural gas burners for laboratory, pilot-, and demonstration-scale tests
- Test the high-heat transfer burner in one of IGT's laboratory-scale high-temperature furnaces
- Design and demonstrate the high-heat transfer burner on IGT's unique pilot-scale glass tank simulator
- Complete one long term demonstration test of this burner technology on an Owens Corning full-scale industrial glass melting furnace
- Prepare an Industrial Adoption Plan. This Plan will be updated in each program Phase as additional information becomes available. The Plan will include technical and economic analyses, energy savings and waste reduction predictions, evaluation of environmental effects, and outline issues concerning manufacturing, marketing, and financing. Combustion Tec, Owens Corning, and IGT will all take active roles in defining this Plan.

During Phase I, the first three objectives were addressed and completed along with the design component of the fourth objective. Work for Phase I was divided in five specific Tasks. A detailed description of each of these Tasks along with a complete description of the results of Phase I efforts is presented below. In the program, IGT is the Prime Contractor and is responsible for meeting project schedule, cost and technical objectives and reporting requirements. Project tasks are designed to be highly interactive, ensuring focused development and rapid commercialization of a commercially attractive burner technology.

Phase I. Research and Development Definition

In Phase I, current glass industry problems and concepts to improve energy efficiency and increase productivity will be characterized. Factors affecting industrial adoption of the high-heat transfer low-NO_x natural gas combustion system will be examined and described. Parametric and thermodynamic descriptions of combustion, soot nucleation and growth, and heat transfer issues will be developed. The burner for Phase II testing will be designed, and the test plans for Phase II and Phase III will be prepared. This Phase is scheduled to require 6 months to complete after project initiation.

Task 1. Characterization of Process Problems and Improvements

The objective of this Task is to prepare a comprehensive description of efficiency, heat transfer, and emissions problems associated with glass melting furnaces. Work by the project team will include literature review, discussions with glass manufacturers, and compilation of data to be used in the Task 2 modeling work. Efforts will be directed toward achieving a detailed understanding of problems in glass production. Problems include the need to increase energy efficiency in oxy-natural gas flames by means such as adding soot to increase luminosity and radiative heat transfer, the need to increase production rate while maintaining glass quality, and the need to reduce pollutant emissions, especially NO_x.

Task 2. Furnace Modeling and Burner Design

Work in this task will begin with development of parametric, heat transfer, and thermodynamic descriptions of the high-heat transfer low-NO_x natural gas burner and the burner as installed in a glass melting furnace. Results of this modeling effort will then be used to prepare a design of a high-heat transfer low-NO_x natural gas burner for the laboratory- and pilot-scale tests in Phase II. Parametric studies and modeling work will be managed by IGT with active Combustion Tec and Owens Corning participation, and the design will be developed by Combustion Tec, Inc. with input from IGT and Owens Corning. A description of the concept behind this innovative burner and the method used to achieve high luminosity by soot formation in the flame is presented in Chapter 1. This compact burner can replace the burners traditionally used in glass melting furnaces with little alteration to the furnace. Combustion Tec is the primary burner supplier to the U.S. glass industry with many years of burner manufacturing experience and extensive business contacts throughout the U.S. glass industry. They will modify the high-luminosity burner design in conjunction with IGT and review by Owens Corning. If more cost-effective manufacturing processes or better mixing methods are identified, they will be adopted for actual production.

Commercially available models to describe oxygen-methane combustion and soot formation and growth will be purchased and modified. The combination of these models will allow prediction of major combustion species, temperatures, and the conditions and levels of soot formation. The combined combustion and soot formation models will be incorporated into more general fluid mechanics and flame radiation models. The parametric and thermodynamic models will solve for mass, species, momentum, and energy balances for turbulent flow and will be validated with laboratory- and pilot-scale testing in Phase II. Adjustments based on experimental results will be made to the models as necessary to accurately describe the burner and the glass melt furnace.

The effort here is to generate a simple kinetics model of soot formation in gas/oxygen flames. Again, a complex model⁶ and simple models⁹⁻¹⁰ of soot formation have appeared in the

literature. In the complex model, the initial soot nucleus (a radical) is "grown" by a lengthy series of elementary reactions to build the "final soot nucleus" (pyrene or acepyrene). This molecule is assumed to "dimerize", "trimerize", etc., by stacking the pyrene plate-like molecules on top of each other. Concurrent with the polymerization reactions (coagulation), the growing soot particle can be oxidized by OH and O₂ to CO or CO₂ in the flame.

In the simple models, the nucleation steps of soot formation are eliminated and empirical correlations of the soot volume fraction as a function of "mixture strength" are used to obtain the number density of soot particles. Problems with this approach are: 1) the correlation is fuel and oxidizer specific and 2) not all mixture strengths are amenable to a single correlation (e.g., very fuel rich or very fuel lean).

All soot models, simple or complex, embody soot coagulation and oxidation steps to arrive at reasonable soot particle sizes compared with experimental values.¹⁸⁻²³ Therefore, we will concentrate our efforts at developing a model which uses a soot nuclei concentration profile (i.e., C₃H₃+) as input and add a few kinetics steps to produce a large soot nucleus - pyrene or coronene sized. Then, we will add reactions for this larger soot nucleus to coagulate and oxidize. The soot model will be compared to literature data on premixed and diffusion flames of methane/oxygen and methane/air, as well as the experimental data acquired in this project for validation and refinement of the model.

The radiation heat transfer model for an oxygen combustion system will rely on an analysis predicting radiative heat transfer from isothermal gas volumes (planar layers and infinitely long cylinders) and isothermal gas and soot mixtures. Concentrations of CO, CO₂, H₂O, and soot will be assumed, and emissivities will then be calculated for different size systems. The completed model will be prepared with protocols to get approximate radiant heat transfer rates as a function of the concentrations of the radiating species. This work will include a scheme for integration over the spectrum. A solution means will be developed to solve the RTE equations for three dimensional systems and account for the spectral nature of emissions and absorptions of radiation.

Using the results of the modeling efforts, the project team (IGT, Combustion Tec, and Owens Corning) will select an optimum burner design for fabrication and use in Phase II testing. Other designs, however, will be prepared in this Task. These other designs will be used if the initial design does not provide sufficient mixing, soot formation, luminosity increase, or improvement in energy efficiency.

Task 3. Phase II and Phase III Research Plan Development

The objective of this Task is to develop the laboratory- and pilot-scale testing plans for Phase II and the demonstration-scale testing plan for Phase III. All of the experimental work in this program will be designed to verify the increase in energy efficiency (greater than 20%), the increase in production rate, and the decrease in NO_x production which result from the use of this technology on a glass melt furnace. All tests will be conducted in oxygen-fired or enriched air-fired furnaces. Testing in an oxygen-fired furnace is preferred because the greatest benefits from this technology will be realized in oxygen-fired furnaces. Laboratory- and pilot-scale tests will be conducted by the project team in IGT facilities with burner fabrication performed by Combustion Tec. The demonstration test in Phase III will be conducted by the project team (IGT, Combustion Tec, Owens Corning) at a commercial glass melting furnace owned by Owens Corning.

Laboratory-scale tests (Phase II) will consist of testing a single high-heat transfer low- NO_x natural gas burner on an existing IGT-owned furnace. A series of tests outlined in Task 6 will be conducted at various conditions of velocity, temperature, stoichiometric ratio, and flow (natural gas and oxygen). Measurements will include pressure, temperatures, furnace load, gas emissions, soot yield and particle size, and radiative and convective heat transfer.

Pilot-scale tests (Phase II) will be conducted in IGT's unique glass tank simulator. A series of tests will be conducted to confirm the burner operating results of the laboratory-scale tests and determine the optimum conditions for the Phase III demonstration test. Pilot-scale tests outlined in Task 7 will be conducted as an oxygen-fired or oxygen-enriched air-fired flame with test variables including velocity, preheat temperature, stoichiometric ratio, and flow (natural gas and oxygen). Gas emissions, heat flux level, furnace temperatures, and flame characteristics will be monitored.

The Phase III demonstration-scale test will be conducted by the project team on a commercial Owens Corning glass melt furnace. The furnace may be oxygen-fired, oxygen-enriched-air fired, or air-fired since this innovative burner is capable of increased energy efficiency under all oxidant-firing conditions. An oxygen-fired furnace is preferred for testing in order to demonstrate the benefits of this technology against the best available high efficiency/low NO_x technology. Conditions will be selected based on the results of laboratory- and pilot-scale tests in Phase II. Independent variables will be controlled in the demonstration test to allow detailed determination of the improved energy efficiency, increase in production rate, and decrease in NO_x emissions.

Experimental testing in Phase II and Phase III will be conducted in existing IGT laboratory- and pilot-scale furnaces and in a commercial Owens Corning glass melting furnace. All project team members will take part in planning the test matrices, preparing burners and furnaces for testing, performing tests, and analyzing test results.

Task 4. Industrial Adoption Plan

An outline of an Industrial Adoption Plan for the high-heat transfer low- NO_x natural gas burner will be developed in this Task. The objective of developing an adoption plan in Phase I of the program is to provide a structured approach to the commercialization of the technology. Update of the Plan through all three Phases of the program will allow test results, design modifications, and industrial concerns to be recognized and managed in a timely manner. The goal of this program is to develop a technology that increases production, improves energy efficiency by at least 20%, and penetrates a significant fraction of the U.S. glass industry. All members of the team recognize this new technology will not be adopted by industry unless the costs, benefits, and potential risks are known, addressed, and then clearly communicated to the end-users. This is the role of the Industrial Adoption Plan.

The Industrial Adoption Plan will include technical and economic analysis, energy savings and waste reduction predictions, an evaluation of environmental effects, and manufacturing, marketing, and financing options. In this Task only an outline of the Plan will be developed. Further details as well as a step-by-step adoption Plan will be included as more input becomes available during Phase II and III of the program.

Combustion Tec will lead in the Industrial Adoption Plan preparation with constant communication and input from the Owens Corning and IGT. In Phase II and III comments

received from additional glass makers will be incorporated into the Plan in order to address specific concerns and streamline the commercialization process.

Task 5. Project Management

This management task will be active throughout Phase I of the program. Management activities will include communication between the team members, co-ordination of all project-related activities, and reporting of project results in a timely manner. Reports for this Phase of the project will be prepared by all team members and will include monthly status reports and a Final report for Phase I. The Final Report will be complete and written to provide the reader with an accurate understanding of the state of development of the high-heat transfer low-NO_x burner system. Included in the Final Report will be parametric and thermodynamic studies of combustion, soot nucleation and growth, an heat transfer as well as the burner design for laboratory testing. A complete discussion will also be included concerning glass industry problems and concepts to resolve these problems and increase energy efficiency and productivity. The initial Industrial Adoption Plan will also be included in the Phase I Final Report.

3.0 Discussion

3.1 Characterization of Process Problems and Improvements

A project review meeting was held at IGT's Energy Development Center on April 18, 1996. Prior to this meeting, IGT and CTI conducted a review of glass furnace operation and characterized process problems and improvements. The results of this review were presented in written and oral form to DOE personnel at this meeting. A summary of the information presented at that project review meeting is presented below.

A review of current technology revealed that the largest problems facing high temperature combustion processes are low productivity, high fuel consumption, and production of high levels of pollutants such as NO_x. All of these issues can be addressed by switching from air-gas firing to oxy-gas firing. However, in most industrial applications, oxy-gas firing is not economically attractive because of the added cost of oxygen. An oxy-gas flame in a hot furnace tends to be non-luminous and provide low radiation heat transfer to the load. The use of oxygen significantly lowers NO_x production, but NO_x is still generated from nitrogen present in the fuel or oxidant or through infiltration.

The solution to the deficiencies with oxy-fuel combustion can be addressed by developing an economical, easily-operated combustion system which increases radiative heat transfer and lowers NO_x emissions. This can be accomplished by seeding the flame with soot particles to generate a highly luminous flame. Higher luminosity increases radiative heat transfer and decreases flame temperature. Lowering the flame temperature lowers the production of NO_x. Increased radiative heat transfer to the load improves the thermal efficiency of the burner and reduces the fuel and oxidant requirements per ton of product melted. This improves the economics of oxy-gas combustion.

The High Luminosity Burner being developed in this project addresses all of the deficiencies identified with oxy-gas combustion. The project team believes that development and application of this new burner will hasten the adoption of oxy-gas firing for high temperature combustion processes. Compared with current oxy-gas combustion systems, the High Luminosity Burner offers higher production rate, an efficiency improvement of a least 20 percent, lower fuel and oxygen consumption, and a significant reduction in NO_x emissions.

Air-gas fired glass furnaces typically rely on regenerators to recover much of the process heat leaving the furnace with the exhaust gas. The amount of exhaust gas is reduced by more than 70 percent by switching to oxy-gas firing. Oxy-gas fired glass furnaces typically do not rely on regenerators to recover process heat. Ideal combustion, with methane representing natural gas, can be described as follows:

Air-Gas Combustion

Oxy-Gas Combustion

Air-gas combustion has several advantages including:

- air is free
- air is safer and simpler to handle
- air-gas combustion processes are well understood with a long record of industrial use

There are also a number of disadvantages associated with air-gas combustion. These include:

- high heat loss with heated nitrogen
- regenerators are needed for heat recovery
- gas velocities are high
- particulate carry-over can be high
- high levels of NO_x formation
- large fans, ducts, and gas cleaning equipment are needed.

The use of regenerators to recover heat from air-gas combustion exhaust gases provides a significant increase in thermal efficiency for these furnaces. A major drawback of the use of regenerators is an increase in NO_x emissions. Figure 3 shows the general air-gas combustion trend for NO_x emissions as a function of combustion air temperature. NO_x emissions increase at a more than linear rate as combustion air temperature increases. This relationship does not affect oxy-gas combustion because the oxygen is not preheated and because there is a low nitrogen concentration in the oxidant.

Figure 3. THE GENERAL EFFECT OF AIR PREHEATING ON NO_x EMISSIONS

Some of the major advantages of oxy-gas combustion are:

- higher combustion temperature

- reduced flue gas volume and velocity
- lower particulates carry-over
- reduced flue gas cleaning costs
- regenerators are not needed for heat recovery
- lower fuel gas consumption
- lower NO_x production
- smaller fans and ducts are required
- water content in glass can be increased

Figure 4 shows the general effect of oxygen enrichment on the process efficiency for a glass furnace. Values will vary from furnace to furnace to the two illustrated trends always hold true. First, increasing the oxygen concentration, at any given flue gas temperature, produces an increase in process efficiency. Second, the impact of flue gas temperature decreases with increasing oxygen enrichment. This figure illustrates the large increase in efficiency provided by regenerators for air-gas fired furnaces. However, as the amount of nitrogen in the gas decreases, the benefit gained with regenerators also declines. For oxy-gas combustion, a regenerator does not provide a very large increase in process efficiency.

Figure 4. THE EFFECT OF OXYGEN ENRICHMENT ON PROCESS EFFICIENCY

In the effort to improve process efficiency and lower NO_x emissions without increasing process expense, the use of oxygen-enriched air as an oxidant has been considered. There is, unfortunately, a large NO_x production penalty realized when oxygen-enriched air is used. Figure

5 shows the general effect of increasing oxygen concentration on NO_x emissions. Enriching the oxidant with oxygen produces higher combustion temperatures. This leads to higher NO_x production as long as large amounts of nitrogen are available for oxidation. Oxygen concentration must be raised to more than 90 percent to achieve the low NO_x production benefits of oxy-gas combustion.

Figure 5. THE EFFECT OF OXYGEN ENRICHMENT ON NO_x EMISSIONS

The High Luminosity Burner being developed and demonstrated in this project is expected to produce significantly lower levels of NO_x than other oxy-gas burners. This is because the High Luminosity Burner will have a significantly lower flame temperature and will therefore generate less NO_x at the same oxygen concentration concentration in the oxidant. This will translate into a cost savings for the High Luminosity Burner because this burner will generate low NO_x using low cost PSA or VPSA oxygen containing 90 to 92 percent oxygen. Other oxy-gas burners will produce higher levels of NO_x , or they must use much more expensive pure oxygen to achieve the same low NO_x emission level.

The temperature and radiant heat flux profiles for air-gas and oxy-gas burners are significantly different. Figure 6 shows that the profiles for air-gas burners are flatter and less peaked than for typical oxy-gas burners. Oxy-gas burners, with high combustion temperatures and peaked profiles, have the potential to provide poor heat distribution and to create hot spots on the furnace crown. The High Luminosity Burner being developed in this program will have a reduced flame temperature and less peaked temperature and radiant heat flux profiles compared with current oxy-gas burners. Also, because the High Luminosity Burner is more efficient than other oxy-gas burners, the amount of fuel and oxygen can be reduced for the same load. This will create even lower temperature and radiant heat flux peaks. These results have been

confirmed by the modeled conducted during Phase I of this project. Detailed results of this modeling are presented in the Topical Report on Modeling.

Figure 6. TEMPERATURE PROFILE COMPARISON FOR AIR-GAS AND OXY-GAS COMBUSTION

A general comparison of some characteristics of oxy-gas and air-gas combustion is presented in Table 2. Oxy-gas combustion has a significantly higher adiabatic flame temperature and a faster normal flame velocity. The speed of oxy-gas combustion initiation is shown by its lower induction time. The higher adiabatic flame temperature leads a decrease in the Arrhenius group (E/RT) and thereby increases the reaction rate coefficient k , equal to $\exp(-E/RT)$.

Table 2. COMPARISON OF OXY-GAS AND AIR-GAS COMBUSTION

<u>Characteristic</u>	<u>Oxy-Gas / Air-Gas</u>
Adiabatic Flame Temperature	1.4
Reaction Rate Coefficient	1.35
Flame Induction Time	0.75
Normal Flame Velocity	1.16

Oxy-gas combustion has a number of advantages over air-gas combustion and has been partially implemented in a number of industries. Some industries using oxy-gas combustion are:

- glass melting
- hearth furnaces (steel industry)
- ladle heating
- aluminum sintering furnaces
- magnesia sintering furnaces
- specialty ceramic production

In all applications, oxy-gas combustion requires special considerations. Specific burners must be designed and used to handle and utilize the characteristics of oxy-gas combustion. For example, burners must operate properly when the laminar gas phase viscosity is higher for oxy-gas combustion while turbulent mixing in the flame is reduced. The oxy-gas conversions in the glass industry have reported 25 to 49 percent reductions in gas usage. Considering the overall cost of glass melting with oxy-gas combustion (including oxygen production), the cost of oxy-gas combustion is no better, and often not as good, as the cost of air-gas combustion with a good regenerator.

The glass industry in the United States was reviewed to assess the possible market for the High Luminosity Burner. Data was taken from an IUPAG report to GRI in Feb. 1995 and from CTI business data. Major points concerning the U.S. glass industry can be summarized by:

- total U.S. glass production in 1992 was more than 81,000 ton/day,
- major segments of the glass industry include container, flat, wool fiber, textile fiber, lighting/TV, press and blown, and sodium silicate,
- container glass represents 55 percent of total glass produced and flat glass represents 20 percent of glass,
- in 1992, 10 percent of the 490 U.S. glass melting furnaces were oxy-gas fired,
- natural gas supplies more than 80 percent of the energy for glass melting,

- natural gas consumption for glass melting is approximately 200 BCF/year,
- NO_x production varies widely between 2 and 15 lb/ton of glass.

A breakdown of glass furnaces by market segment and furnace type is provided in Table 3.

Table 3. COMMERCIAL GLASS MANUFACTURING IN THE UNITED STATES IN 1992

	No. of Plants	Tons per day	Furnaces	Regen. Sideport	Regen. Endport	Unit Furnaces	Oxy/Fuel	All Electric
Container	68	48,000	154	68	66	--	13	7
Flat	29	19,000	38	36	--	--	1	1
Wool Fiber	24	1,400	58	--	--	16	5	37
Textile Fiber	12	2,100	60	--	--	47	6	7
TV/lighting	19	3,000	60	19	--	30	9	2
Press and Blown	31	3,400	95	32	10	33	11	9
Sodium Silicate	24	4,500	25	20	--	5	--	--
Total	207	81,400	490	175	76	131	45	63

CTI is the leading supplier of burners and combustion systems to the U.S. glass industry. They have been involved in the trend toward oxy-gas combustion and are actively seeking new products to introduce to the market. The move toward oxy-gas combustion is expected to continue and accelerate when improved combustion system such as the High Luminosity Burner become available. CTI has observed and participated in the conversion to oxy-gas combustion and has observed the following historical trend:

- 1970s
 - Oxygen enrichment was used for productivity increase on some furnaces.
 - Some small, specialty glass furnaces (less than 50 ton/day) were converted to oxy-gas.
- 1980s
 - Some fiberglass furnaces converted to oxy-gas.
 - A few container glass furnaces converted to oxy-gas.
- 1990s.
 - Most fiberglass and specialty glass furnaces converted to oxy-gas.
 - Half of container glass furnaces converted to oxy-gas.
 - All TV/lighting glass furnaces converted to oxy-gas.
 - One or two flat glass furnaces converted to oxy-gas.
- 2000
 - Approximately 20 percent of all glass furnaces will be converted to oxy-gas.

CTI has converted furnaces in the container, fiberglass, frit, TV, lighting, and tubing markets from air-gas firing to oxy-gas firing. These conversions have been completed for more than sixteen companies and total more than forty furnaces. Different industrial situations lead to

various reasons to adopt oxy-gas combustion in a glass furnace. Figure 7 shows the leading reasons for conversion from air-gas to oxy-gas firing. A reduction in NO_x emissions is clearly the dominant reason for switching from air to oxygen. Other important reasons include a reduction in capital costs, fuel savings, an increase in production, and a decrease in particulates. Several other reasons that apply to a few furnaces include an increase in glass quality, the ability to reduce electric boost and save on the electricity costs, and a savings on batch to the furnace.

Figure 7. REASONS FOR CONVERTING GLASS FURNACES TO OXY-GAS FIRING, PERCENT OF FURNACES

Figure 7 confirms that the dominant reason to convert a furnace to oxy-gas combustion or to build a new oxy-gas furnace is to reduce NO_x emissions. Conversion rates have been low because glass makers have not had compelling economic reasons to convert to oxy-gas firing. Figures 8 and 9 demonstrate that the economics of oxy-gas combustion are improving. Over a four year period from 1991 to 1995, natural gas prices have oscillated and electricity costs have remained unchanged. During this same period, oxygen production technology has improved, and oxygen costs have decreased by as much as 30 percent. Also during this time, a number of oxy-gas conversions were completed, and industry became more familiar and comfortable with this new melting technology. Oxygen companies and burner companies (such as CTI) have aggressively marketed oxy-gas combustion systems to the glass industry.

Figure 8. CHANGES IN ENERGY AND OXYGEN PRICES

Figure 9. OXY-FUEL COSTS AND BENEFITS BASED ON A 300 TON/DAY CONTAINER GLASS FURNACE REBUILD

The cost analysis in Figure 9 is a general breakdown for the rebuild of a large container glass furnace. The analysis shows that the cost of oxygen outweighs any single benefit of the

conversion. However, when all the benefits are considered, they can outweigh the cost of oxygen. Also, as the cost of oxygen decreases, this cost-benefit analysis looks more attractive to the glass maker.

The project team studied the commercial oxy-gas burners that are currently available. All of these burners use one of the following methods to achieve low NO_x emissions.

- reducing the peak flame temperature
 - temperature can be reduced by entraining furnace gases
 - delayed mixing can also reduce temperature
- increasing flame area
 - a low momentum flame covers more area
 - a flat flame, with greater width than height, also covers more area
- increasing flame luminosity
 - staging combustion can increase luminosity
 - soot burnout increase luminosity. Precombustion can be used to generate soot.

Table 4 provides a list of the commercially available high luminosity, low NO_x oxy-gas burners for glass furnaces. All manufacturers claim their burners provide high heat transfer and low NO_x emissions. Operating characteristics, however, vary widely among the commercial burners, and the High Luminosity burner is expected to provide at least 20 percent higher heat transfer to the load than the best currently available burner while producing less NO_x.

Table 4. COMMERCIAL OXY-GAS BURNERS CLAIMING HIGH LUMINOSITY AND LOW NO_x

<u>Company</u>	<u>Burner</u>	<u>Method</u>
AGA Air Products BOC Gases	OXY-GAS CLEANFIRE Flat Jet	low velocity (momentum) cracking via precombustion, delayed mixing low initial mixing rates allow fuel pyrolysis and soot particle nucleation; flame stabilized close to burner to obtain laminar fuel flow; rectangular flow
Combustion Tec, Inc.	Primefire 100	low momentum, precombustion for soot formation
Combustion Tec, Inc.	Primefire 300	low momentum, precombustion for soot formation, flat flame
Maxon Corp.	LE	staged O ₂ with holes through burner block to achieve low NO _x
Union Carbide	'A' Burner	dilution of oxygen via aspiration of furnace gases to reduce peak flame temperature; smaller oxygen flow to annulus around natural gas for stability
Xothermic	--	conventional oxy-gas burner

A number of other oxy-gas burners have also been identified. A list of these burners is presented in Table 5. These burners are either older models or not as well known as the burners described in Table 4.

Table 5. OTHER OXY-FUEL BURNERS

<u>Company</u>	<u>Burner</u>	<u>Claim</u>
Air Liquide	ALGLASS	Low NO _x
Airco	---	--
Air Products and Combustion Tec	KT-3	--
Empco	---	--
Kaiser Marquardt	---	high luminosity
Korf	---	--
Maxon Corp.	Kinemax	low NO _x
Maxon Corp.	OXY-THERM	low momentum bushy flame
North American	---	high luminosity
Praxair, Inc.	---	high luminosity

The characterization study was conducted by CTI with input from IGT. The conclusion is that a sizable, and growing, market exists for the High Luminosity Burner, being developed in the program. Strong demand, based on economics and emissions savings, is anticipated in the glass industry and in other markets. During Phase II, laboratory and pilot-scale High Luminosity Burner prototypes will be tested and evaluated relative to the best commercial oxy-gas burners.

3.2 Modeling

A Topical Report presenting modeling study results for the novel high-heat transfer low-NO_x natural gas combustion system being developed, tested, and demonstrated in this program has been prepared. This Topical Report on Modeling is being submitted to DOE in partial fulfillment of the Phase I contractual requirements. Before construction and testing of prototype burners, the Novel High-Heat Transfer, Low-NO_x Natural Gas Combustion System was mathematically modeled to verify the predicted improvements in heat transfer and lower NO_x compared with conventional burner systems. The burner system, referred to as the High Luminosity Burner, is described by the modelers as being composed of a Fuel Preheating Zone and a Flame Zone. The Fuel Preheating Zone was modeled by a group at the University of Illinois at Chicago headed by Prof. Alexander Fridman, and the Flame Zone was modeled by a team at Purdue University headed by Prof. Raymond Viskanta.

Modeling was conducted for the High Luminosity Burner system firing into an oxy-gas glass melting furnace. This application of the burner will be demonstrated in Phase III of the project. Many other oxy-gas and air-gas fired applications of this burner system have been identified by the project team but have not yet been modeled. This work is important to the development of the High Luminosity Burner but is outside the scope of the current project.

The Topical Report on Modeling includes the results from both modeling teams. A Summary is provided along with Appendices presenting reports from the modeling teams with

detailed results for both the Fuel Preheating Zone and the Flame Zone of the High Luminosity Burner. The Summary includes the High Luminosity Burner concept, the modeling strategy, results and discussion of modeling efforts, and conclusions. The discussion of the Fuel Preheating Zone modeling covers natural gas preheating; pyrolysis and oxidation reactions and reaction pathways; and soot precursor and soot nucleation, growth, coagulation, and oxidation in a laminar, preheated methane-oxygen flame. Results are presented in Appendix 1 of the Topical Report.

Modeling of the Flame Zone began with development of an idealized, one dimensional (1-D) model for predicting temperatures and radiation heat transfer to the load from non-luminous flames and from flames containing soot. The impact of axial heat transfer was then determined through the development of a quasi-two dimensional (quasi 2-D) model of the Flame Zone and combustion space. Results obtained using the quasi 2-D model are consistent with the 1-D modeling results and are presented in Appendix 2 of the Topical Report. The Appendix describes and discusses parametric calculations which show the increase in radiation heat transfer to the load resulting from the presence of soot in the products of combustion.

Kinetic, heat transfer, and thermodynamic modeling of both Zones, Fuel Preheat and Combustion, of the High Luminosity Burner system were completed. The modeling results for the Fuel Preheat Zone were provided as an input to the Combustion Zone modeling. Results were extremely encouraging for the High Luminosity Burner concept. Major conclusions of the modeling studies are:

- Only 2 to 9% of the feed natural gas must be combusted to provide heat for soot formation.
- Short residence times are required (10^{-2} to 1 seconds at 1300 K and 10^{-4} to 10^{-2} seconds at 1600 K) to produce soot precursors and soot.
- Soot concentration increase from 0.1 to 2.0 volume percent when optimum concentrations of soot precursors are formed in the Fuel Preheat Zone.
- Soot concentrations in the fuel and at the furnace inlet are important parameters affecting furnace thermal performance.
- At high enough soot concentrations, the thermal efficiency of luminous flames can be increased by 25% with no changes in other parameters.
- Flames covering a larger fraction of the glass surface enhance the thermal performance of the furnace.
- Short flames are undesirable because they produce highly peaked and non-uniform profiles at the glass surface and create high gas and refractory temperatures.
- Increasing soot concentration decreases gas temperature which is expected to lower NO_x formation.
- Crown temperatures are determined to increase slightly with the addition of soot to an oxy-gas flame.

The results of the modeling effort have been incorporated into the design of the prototype High Luminosity Burner which will be fabricated and tested during Phase II of this DOE program. During Phase II, testing results will be compared with the modeling predictions. Any discrepancies between modeling and practice will be examined before proceeding to the next

level of testing. Phase III work will involve demonstration of the burner system on a commercial oxy-gas glass furnace.

3.3 Burner Design

After the modeling work was completed, engineers at CTI designed a prototype High Luminosity Burner for laboratory testing on an IGT furnace. Complete CAD drawings of the first prototype burner have been prepared along with instructions for fabrication. The design team included all components of the burner system in the Fuel Preheat Zone and the Flame Zone. The sections included in the two furnace Zones are:

- Fuel Preheat Zone
 - Precombustor
 - Indirect natural gas heating
 - Direct natural gas heating
 - Mixing
- Flame Zone
 - Fuel-rich combustion
 - Fuel-lean combustion

The first prototype burner is designed to operate at a maximum capacity of 0.5 MM Btu/h with a 4:1 turndown. A large amount of flexibility is designed into this burner to allow for variations in temperatures, residence times, mixing characteristics, and flame section distributions and oxygen/fuel ratios. The burner will also be equipped with test ports to allow collection of gas samples for analysis, to measure temperatures, and to observe the flame. The burner concept is proprietary to IGT, and detailed discussions of the exact burner configuration are outside the scope of this project. Therefore, drawings and detailed descriptions are not included in this report.

During Phase II of this project, a prototype burner will be fabricated by CTI from the design developed in Phase I. This burner will be tested by IGT, and changes and improvements will be incorporated in a new design. This new design will be used to fabricate a 3 MM Btu/h High Luminosity Burner that will be tested on an IGT pilot-scale furnace. This second burner will be used to compare the High Luminosity Burner system with commercial oxy-gas burners. After completing this testing in Phase II, the project team will fabricate full-scale burners (at CTI) and demonstrate the new combustion system on a commercial Owens Corning oxy-gas glass furnace.

3.4 Phase II Research Plan

Phase II will include laboratory-scale testing of the high-heat transfer low-NO_x natural gas burner system and pilot-scale testing of the system on a glass tank simulator. Engineering design and prototype testing will be incorporated into the laboratory- and pilot-scale testing

campaigns. All project team members will participate in the development of test plan matrices, preparations for testing, testing, and analysis of the test results. An outline of the Phase II testing schedule was presented in Task 6 and Task 7 of the Work Plan in the project proposal. The testing schedule presented below is similar to the originally proposed test schedule. A substitution of a different high temperature furnace instead of the glass tank simulator may be made for pilot-scale testing. This substitution, if made, would have no impact on the conditions of testing but would allow more tests to be made with more burners. The test plan is described in two parts: laboratory-scale testing and pilot-scale testing.

3.4.1 Laboratory-Scale Testing

The prototype High Luminosity Burner designed by Combustion Tec during Phase I will be fabricated by CTI, delivered to IGT, and installed on the laboratory-scale furnace for testing. The prototype burner has a firing capacity of 0.5 MM Btu/h with a 4:1 turndown ratio. Details of the burner design are proprietary, but a general description is provided in the Burner Design section of this report. Based on testing results with the first burner and available resources, the project team may choose to fabricate and test a second, 0.5 MM Btu/h prototype burner.

The furnace to be used for laboratory-scale testing of the High Luminosity Burner is a well instrumented test furnace located in IGT's combustion laboratory. This flexible test unit has a firing rate of up to 500,000 Btu/h and is fully equipped with controls and measurement devices. A schematic diagram of the unit is presented in Figure 10. Modifications for testing in this program will include providing oxygen service, changing burners, piping and metering the natural gas and oxygen, installing and calibrating the gas analysis equipment, and preparing or removing the furnace load as desired. This work will be performed by IGT personnel.

Figure 10. LABORATORY-SCALE TEST FURNACE

The laboratory-scale furnace is 7 feet long with a 15 in. by 15 in. cross section. The furnace is lined with 3000°F fiberboard insulation. Up to sixteen water-cooled tubes can be lowered into the furnace to provide variable loads. An available, electric preheater can be used to preheat the natural gas. Nine ports on both sides of the furnace and three ports at the back end provide for gas sample, temperature, and optical data collection. The front end is removable to allow for installation of many types of burners with capacities up to 1 MM Btu/h. The top of the furnace is removable to allow access to the furnace chamber. Natural gas, air, oxygen, and load water flow rates and temperatures are measured. Gas analyzers are available for measuring the NO_x , CO, CO_2 , O_2 , and THC concentrations in the combustion gases.

Laboratory-scale tests will include tests with a standard burner, tests with the High Luminosity Burner with no Fuel Preheating Zone, and tests with the High Luminosity Burner over a range of conditions. All tests will be oxygen-fired: no air-fired or enriched air-fired tests are planned. Natural gas will not be preheated before entering the Fuel Preheat Zone of the burner. A test campaign consisting of up to 150 test conditions is scheduled. Independent test variables and their ranges are outlined in Table 6. Multiple test points can be covered in a single test period, and sufficient test duplication (at least 25%) will be included to verify the data precision. All instruments will be calibrated regularly to assure data accuracy.

Table 6. LABORATORY-SCALE TEST CONDITIONS

<u>Test Burner</u>	<u>Independent Operating Variable</u>	<u>Range of Test Conditions</u>
Standard Burner - Baseline Tests	Firing Rate, Btu/h	125,000 - 500,000
	Overall Excess O ₂ , %	0 - 10
High Luminosity Burner	Firing Rate, Btu/h	125,000 - 500,000
	Overall Excess O ₂ , %	0 - 10
	Fuel Preheat Fraction, %	0 - 12
	Preheat Residence Time, sec.	0 - 1 sec.
	Mixing Geometry	various
	First Stage Combustion O ₂ /Gas Stoichiometry	0.7 - 0.9
	First Stage Combustion Residence Time, sec.	various

The laboratory-scale High Luminosity Burner is designed to have complete flexibility for controlling natural gas and oxygen flows. Varying these flows and changing the mixing section length and geometry will allow study of all independent parameters of concern. The prototype burner will also have internal thermocouples and a gas sampling port to provide for evaluation of operating results. Test measurements will allow determination of efficiency improvements and NO_x reduction levels related to test conditions. Measured values in the laboratory-scale tests will include:

- O₂, CO₂, CO, total hydrocarbons (THC), and NO_x in the flue and at selected points along the flame using water cooled probes and a continuous emissions monitoring system (CEMS)
- Fuel preheat temperature
- Heat flux profile along the furnace length using adjustable water-cooled load and heat flux sensors
- Total heat flux to the load
- Total and radiative heat flux measured by sensors
- Furnace refractory temperatures using embedded thermocouples to obtain a temperature profile.
- Exhaust gas temperature
- Flame luminosity measured optically
- Flame temperature measured with an optical pyrometer

- Burner wall temperatures
- Gas temperatures in the furnace using suction pyrometers
- Flame length
- Gas composition including soot precursors in the mixing section using grab samples.

Data collected in laboratory-scale testing will be analyzed to determine optimum burner operating conditions. These conditions include fraction of gas burned in the preheater, preheater residence time and exit temperature, oxygen-natural gas stoichiometric ratio and residence time in the first, fuel rich combustion stage, firing rate, and overall level of excess air for the entire burner. Results of the analysis will be used to design and fabricate the larger burner for pilot-scale testing. Information gained from testing in the laboratory-scale furnace will also be considered during modification of the Industrial Adoption Plan in Phase II. All team members will participate in data analysis and share findings, results, and conclusions.

3.4.2 Pilot-Scale Testing

A pilot-scale High Luminosity Burner will be fabricated by CTI with input from the other team members. The burner will be delivered to IGT for installation and testing on the pilot-scale glass tank simulator. The burner will be based on the design of the prototype laboratory-scale burner developed in Phase I and test results obtained with the laboratory-scale burner. The pilot-scale burner is anticipated to have a firing rate of 2 MM Btu/h with a 4:1 turndown ratio. This is equivalent to the smallest size of a commercial oxy-gas burner for glass furnaces. A high degree of flexibility will be built into the pilot-scale burner to allow for adjustments of natural gas and oxygen flow rates, to permit changes in preheat length to control preheat residence time, and to provide variable first stage combustion oxygen injector geometry to change first stage combustion residence time. A single test burner will be fabricated for this project, but a second burner may be built and tested if design limitations in the first burner prevent optimum burner performance.

The pilot-scale furnace to be used for burner testing is an existing IGT glass tank simulator with a firing capacity of 2 MM Btu/h. A schematic diagram of this full instrumented test facility is presented in Figure 11. Modifications for testing in this project will include changing burners, piping and metering the natural gas and oxygen, and installing and calibrating the gas analysis equipment. This work will be performed by IGT personnel. The project team may elect, with the approval of DOE, to substitute a second IGT burner test facility for the glass tank simulator. This second furnace has the same firing capacity, temperature capability, and measurement options. This facility is easier and less expensive to operate and would therefore allow the project team to conduct more pilot-scale High Luminosity Burner tests. Whichever test facility is used, the same test matrix will be followed. No furnace substitutions will be made without DOE approval.

Figure 11. PILOT-SCALE GLASS TANK SIMULATOR

The pilot-scale glass tank simulator has internal dimensions of 3 feet in height, 4.5 feet in width, and 11 feet in length is insulated with 3200°F hard refractory. The furnace dimensions were chosen to simulate a section of the combustion space of a commercial glass tank. To simulate glass tank conditions, a special heat sink consisting of a 1.5 inch top layer of molten glass (during operation) is supported by a 1 inch layer of refractory bricks which are cooled at the bottom by a series of water-cooled heat exchangers.

Oxygen preheating will not be used for testing in this project. However, combustion air for the furnace can be heated to 1200°F in an indirect-fired air heater and then to about 2200°F in a direct-fired air heater. Oxygen is added to make up for the oxygen losses in the direct-fired heater. The simulator is equipped with a flat-floored port scaled to fit the furnace. The flat-floored port was design for 45° and 90° side-of-port firing and 20° to 50° overport firing. Various burners, including Combustion Tec variable orifice gas burners, have been used in the glass tank simulator.

The furnace has 30 removable panels along one sidewall for insertion of gas sampling, heat flux, and temperature measurement probes. Thermocouples are used to monitor the internal refractory surface temperatures. The refractory-shielded thermocouples are installed so that they tips are flush with the internal surface of the roof and the sidewall. Gas analysis equipment for measuring the O₂, CO, CO₂, NO_x, and THC concentrations in the exit port are available in the system. The test facility has recently been equipped with a computerized system for continuous data monitoring and acquisition.

Furnace modifications required for this project include removing the variable orifice burner and installing the High Luminosity Burner, changing the feed gas system to work with

oxygen instead of preheated air, and connecting and calibrating all of the test measurement instrumentation.

A series of up to 20 pilot-scale tests will be conducted by IGT-led project team staff in the existing glass tank simulator. A second furnace capable of the same operating conditions may be substituted, with DOE approval, if this would allow for additional testing. Combustion Tec will fabricate the burner. Combustion Tec and Owens Corning will participate in the test matrix design, testing, and data analysis. Heat transfer to the load and NO_x emission level have been previously found to depend on the burner position and the degree of mixing. A preferred burner position and gas velocity will be selected to optimize energy efficiency and minimize NO_x production.

Pilot-scale tests will include baseline tests with a standard burner and a series of tests with the high-heat transfer low-NO_x natural gas burner. Tests with the High Luminosity Burner will include the case in which the Fuel Preheat Zone is not used. The test matrix will be completed in a single furnace campaign. Oxy-fuel firing will be used for all tests. No air-fired or enriched air-fired tests will be conducted. Natural gas will not be preheated before entering the preheating section of the burner. While a preliminary test matrix consisting of approximately 20 tests is scheduled, duplicate tests will be conducted to verify data precision. Ranges of test conditions are summarized in Table 7.

Table 7. PILOT-SCALE TEST CONDITIONS

<u>Test Burner</u>	<u>Independent Operating Variable</u>	<u>Range of Test Conditions</u>
Standard Burner - Baseline Tests	Firing Rate, Btu/h	500,000 - 2×10^6
	Overall Excess O ₂ , %	0 - 10
High Luminosity Burner	Firing Rate, Btu/h	500,000 - 2×10^6
	Overall Excess O ₂ , %	0 - 10
	Fuel Preheat Fraction, %	0 - 12
	Preheat Residence Time, sec.	0 - 1 sec.
	Mixing Geometry	various
	First Stage Combustion O ₂ /Gas Stoichiometry	0.7 - 0.9
	First Stage Combustion Residence Time, sec.	various

Similar test measurements will be made for the pilot-scale tests as were made for the laboratory-scale tests in Task 6. Test measurements will allow determination of energy efficiency improvements and NO_x reduction levels related to test conditions. Measured values in the pilot-scale tests will include:

- O₂, CO₂, CO, total hydrocarbons (THC), and NO_x in the flue and at selected points along the flame using water cooled probes and a continuous emissions monitoring system (CEMS)
- Fuel preheat temperature
- Heat flux profile along the furnace length using adjustable water-cooled load and heat flux sensors
- Total heat flux to the load
- Total and radiative heat flux measured by sensors
- Furnace refractory and load temperatures using embedded thermocouples to obtain a temperature profile.
- Exhaust gas temperature
- Flame luminosity measured optically
- Flame temperature measured with an optical pyrometer
- Burner wall temperatures
- Gas temperatures in the furnace using suction pyrometers
- Flame length
- Gas composition including soot precursors in the mixing section using grab samples.

All project team members will analyze test data and discuss results and conclusions. Data collected from pilot-scale tests in the glass tank simulator will include heat transfer to the load, NO_x emissions, and temperatures of the exit gas and in the tank. Burner independent variables of natural gas flow and temperature, oxygen flow and temperature, velocity, stoichiometric ratio, etc. will be controlled at desired testing levels. Increases in heat transfer and decreases in NO_x emissions with the high-heat transfer low-NO_x natural gas burner will be calculated with reference to baseline cases fired under similar conditions using conventional burners. Comparisons of various firing conditions will allow selection of the preferred operating conditions for the demonstration test in Phase III.

3.5 Industrial Adoption Plan

Task 4 in Phase I calls for the development of an Industrial Adoption Plan outline. The following outline was prepared by Combustion Tec, the commercial partner in the project team. When the project moves into Phases II and III, this outline will be expanded and targeted based on testing and market information. An Industrial Adoption Plan developed through the project and incorporating the views of a number of industrial sources will streamline the High luminosity Burner commercialization process.

The following outline lists pertinent information necessary for the successful marketing

of a new burner. This information will be obtained or generated as a result of the High Efficiency Low NO_x ("HELN") Burner Development program. Combustion Tec, with the aid of IGT and Owens Corning, will assemble the information into technical product brochures, sales brochures, and presentations. Technical brochures will be supplied to present oxygen-gas glass furnace operators. Technical brochures and presentations will be made to new prospective customers of the technology. Other marketing efforts will include advertisements in trade magazines and presentations at trade shows.

Combustion Tec, Inc. has in the past offered new products on a limited trial basis to increase sales and market penetration. The industrial adoption plan may also include field demonstration of burner(s) and flow control equipment which can be installed at any oxygen-gas fired furnace for the trial period.

I. Technical Information

- A. Burner Operating Theory and Principle - A high temperature natural gas preheating burner with soot formation and subsequent soot burnout in the flame, increasing the burner's energy efficiency and furnace throughput, while minimizing the furnace air emissions.
- B. Burner Firing Capacity (MM Btu/hr) - Prototype lab scale burner will have a maximum firing capacity of 0.5 MM Btu/hr. Data from this prototype will be used to develop a larger development burner and eventually a commercial model at various capacities.
- C. Burner Turn Down Ratio/Range of Burner Firing Capacity - The optimum burner design will offer at least a four to one (4:1) turn down.
- D. Operating Velocities (ft/s) - The prototype burner will offer flexible firing velocities in the range of 50 - 180 ft/s with interchangeable parts for easy replacement and evaluation. These velocity adjustments will be used for fuel oxidation, indirect heating of natural gas, direct heating of natural gas, fuel rich combustion, and remaining oxygen requirements.
- E. Operating Temperatures (°F) - The temperatures for optimum performance will be varied during lab testing for best possible results in the mechanism of soot formation.
- F. Operating Pressures, Gas and Oxygen (PSIG, IWC) - Operating pressures are expected to be in the range of 5 IWC to a maximum of 4.5 PSIG. This operating data will be laboratory verified.

II. Burner Selection

- A. Body Size - The preliminary design for the HELN burner specifies a burner which has an overall length of 3'-8". Gas inlet sizes of 1/4" FNPT for indirect natural gas and 1" MNPT for direct natural gas are specified. Oxygen inlet sizes of 1" MNPT for fuel rich oxygen and 1" MNPT for fuel lean oxygen are also specified.
- B. Tip Sizes - The prototype burner will offer adjustable orifice burner design technology to allow for flexibility and interchangeability.

C. Burner Block - The burner block or precombustor tile will use materials that have been successfully used with Combustion Tec, Inc.'s Primefire® series burners.

III. Operation

A. Installation and Mounting - Installation and mounting methods will be developed based upon Combustion Tec, Inc.'s Primefire® series and traditional air fuel methods.

B. Hose Connections - Quick coupler devices with safety locks will be used to provide easy successful connections to the burner.

C. Flow Controls and Adjustments - Flow control equipment will need to be designed and sized according to supply and usage applications.

D. Velocity Adjustments - Burner velocity adjustments will be made through the use of adjusting lead screws from Combustion Tec, Inc.'s adjustable orifice designs.

E. Optimizing Burner Performance - Throughout the development phase all project partners will participate in optimizing the engineered product.

IV. Manufacturing

A. Materials - Material selection can be broken down into:

i. Burner - Will consist of stainless steels; primarily 304, 316, and 446, as well as some Inconel 600.

ii. Seals - Will consist of viton o-rings or another oxygen compatible material. Temperature will also play an important role in this selection process.

iii. Burner Block (Precombustor) - Will consist of AZS bonded material.

B. Construction Methods - Standard machine shop and design practices will be used in the manufacturing, machining and welding of the burner. Quality checking of each individual part will be done to assure manufacturing tolerances are maintained.

C. Assembly - Two methods will be employed; the first method is to properly oxygen clean the entire burner assembly and the second method is to assemble using oxygen compatible lubricants and seals.

V. Maintenance

A. Cleaning - Maintenance will be evaluated throughout the development phase. Build up of soot deposits within the burner itself is a consideration that will be evaluated. For a successfully marketed product, a low maintenance product is a must.

B. Replacement Parts - Spare parts are always considered when designing a product, for such places where expected wear may take place, or where interchangeability is needed for adaptation.

VI Base Line Comparisons

A. Air/Fuel - Traditional air/fuel burners may be studied to make comparisons.

B. Oxy/Fuel - Testing of the Primefire® series burners would offer a comparison

between today's technology and tomorrow's ultra low NO_x efforts.

VII. Economic Analysis

A. Furnace Efficiency / Improvement - An on-going analysis will be required but preliminary information suggests the following:

Improvements can be quantified in terms of heat energy required to melt a unit ton of glass. In Table 8, a typical regenerative furnace melting efficiency is compared with the current oxy-gas furnace and predictions are made for the HELN burner on the same furnace.

Table 8. MELTING EFFICIENCY COMPARISON

Type of Furnace	MM Btu/Ton	% Change
Regenerative (Sideport)	4.5	-----
Oxy-Fuel (Current)	3.8	15
HELN Oxy-Fuel (Proposed)	3.0	34

Notes:

1. A 300 tons/day container glass furnace is considered for analysis.
2. All furnaces use electric boost for maintaining glass quality.
3. HELN burners provide a 20% improvement in melting efficiency over the current oxy-fuel furnaces.

Table 8 shows that overall efficiency improvement of the HELN Oxy-fuel furnace over the regenerative furnace is significant (~34%).

B. Cost Comparisons - This will require on going analysis throughout the development stage to refine the information available. Preliminary information suggests the following:

Table 9 compares relative melting costs of the furnaces described in Table I translated in cost savings per unit weight of glass produced.

Table 9. MELTING COST COMPARISONS

Type of Furnace	Fuel (SCFH)	Oxygen (SCFH)	Costs (\$/Ton)			% Change
			Fuel	O ₂	Total	
Regenerative (Sideport)	45,000	---	10.80	---	10.80	---
Oxy-Fuel (Current)	38,000	76,000	9.12	9.12	18.24	+69
HELN Oxy-Fuel (Proposed)	30,000	60,000	7.20	7.20	14.40	+33

- Notes:**
1. A 300 tons/day container glass furnace is considered for analysis.
 2. Fuel is natural gas @ \$3.00/MCF and oxygen is at \$0.15/CCF.
 3. Electric boost costs are not considered.
 4. HELN Oxy-fuel is considered to be 20% more fuel efficient than current oxy-fuel.

Table 9 shows that HELN burners will provide significant savings in both fuel and oxygen costs on a ton of glass basis. For example, HELN Oxy-fuel burners will provide annual fuel and oxygen savings in the amount of \$420,480. This amount would pay for the HELN burner hardware in just one year.

C. Return On Investment - Additional advantages of the HELN Oxy-fuel burner include use of lower purity oxygen (such as VPSA oxygen), which would allow glass producers to purchase oxygen at a lower cost and further reduce melting costs. Current VPSA systems produce oxygen at 92% purity, however, the HELN burner would allow "equivalent" operation (both in heat output and emissions), at much lower purity. This number can be lowered to 75% purity, by enhancing the radiative heat output and reducing NO_x formation in the primary flame region. The above mentioned data show lower operating costs for HELN as opposed to traditional oxy-fuel burners. Capital cost for oxy-fuel technology is significantly less than for air-fuel technology, while operating cost increases. The net result is a savings.

VIII. Emission Data

- A. NO_x - The benefits of the HELN Oxy-fuel burner in emission control are significant. This can be shown by NO_x abatement costs as computed in Table 10.

Table 10. NO_x ABATEMENT COSTS

Type of Furnace	Typical NO _x Emissions (Lb./Ton)	NO _x Regulation (Lb./Ton)	Annual NO _x Abatement Costs @ \$3,000/Ton NO _x
Regenerative (Sideport)	5.5	0.5	\$821,250
Oxy-Fuel (current)	1.0	0.5	\$82,125
HELN Oxy-Fuel (proposed)	0.5	0.5	0

- Notes:
1. A 300 tons/day container glass furnace is considered for analysis.
 2. NO_x regulation is set at 0.5 lb./ton for the purpose of example only.
 3. NO_x abatement cost is assumed at \$3,000/ton NO_x for all three cases.

Table 10 shows that the regenerative furnace would require significant expenditure in the form of NO_x abatement costs every year to comply with the NO_x regulations. On the other hand, the HELN Oxy-fuel furnace operating at ultra-low NO_x can accumulate NO_x credits (in the non-attainment area) for future use or outright sale to other energy intensive industries.

B. CO - Further studies as to the products of combustion will be developed throughout the tests in Phase II and Phase III.

C. CO₂ - Further studies as to the products of combustion will be developed throughout the tests in Phase II and Phase III.

IX. Financing

A. Selling Price - Will need to be considered by conducting a market analysis and will be developed in later project Phases.

B. Lease - A lease option may be considered as a cost effective alternative to sale.

4.0 Summary, Conclusions and Recommendations

The two major objectives of Phase I have been successfully completed on time and budget. These include modeling of the high-heat transfer low-NO_x natural gas combustion system and preliminary design of the prototype 0.5 MMBtu/h burner. Modeling work was conducted by the University of Illinois at Chicago (for the preheat zone) and Purdue University (for the flame zone) with supervision by IGT. The modeling results confirmed the >20 percent increase in heat transfer expected with this combustion system. A complete list of modeling study conclusions is presented below. Combustion Tec Inc. design a prototype burner that will be used for laboratory-scale testing during Phase II of this project. The design allows for adjustment of all flows to the burner, changes in direct mixing patterns, and variation in residence time of gases during preheating and mixing in the burner. All Phase I objectives have been met and work will proceed immediately with Phase II.

Since tests have not yet been carried out, the following conclusions are based on the modeling studies:

- Soot concentration increase from 0.1 to 2.0 volume percent when optimum concentrations of soot precursors are formed in the Fuel Preheat Zone.
- Soot concentrations in the fuel and at the furnace inlet are important parameters affecting furnace thermal performance.
- At high enough soot concentrations, the thermal efficiency of luminous flames can be increased by 25% with no changes in other parameters.
- Flames covering a larger fraction of the glass surface enhance the thermal performance of the furnace.
- Short flames are undesirable because they produce highly peaked and non-uniform profiles at the glass surface and create high gas and refractory temperatures.
- Increasing soot concentration decreases gas temperature which is expected to lower NO_x formation.
- Crown temperatures are determined to increase slightly with the addition of soot to an oxy-gas flame.

The project team is confident that the objectives of this project can be met based on the modeling results obtained in Phase I. Therefore, the team recommends proceeding directly with Phase II. Phase II involves detailed design, construction, and laboratory-scale testing of a prototype 0.5 MMBtu/h burner. This first burner will be built to provide as much flexibility as possible in flow adjustments, preheat and mixing residence times, and direct mixing patterns. Following the prototype burner testing, the team recommends continuing with Phase II to design, construct, and test a pilot-scale 2 MMBtu/h high luminosity burner. This burner will be close to a commercial design. This means some flexibility will be built into the burner, but the design will be simplified to approach an operating commercial burner. The team further recommends to compare the heat transfer and emissions characteristics of the laboratory-scale and pilot-scale high luminosity burners with commercial oxy-gas burners evaluated on the same test stands. These recommendations are in agreement with the original test plan for this project. No major adjustments in work scope have been made or recommended at this time.

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APPENDIX A

STIMULATION OF INTERMEDIATE SOOT FORMATION HIGH LUMINOSITY BURNER

University of Illinois, Chicago

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Model

Detailed modeling of soot particles nucleation, growth, coagulation and oxidation in laminar preheated methane-oxygen flames is presented for the case of high luminosity burner. The employed kinetic model was developed on the base of well-known kinetic schemes elaborated and tested earlier for different regimes of methane-air flames. Special attention was paid to kinetic model of soot formation, time evolution of soot particles and soot oxidation on the final stage of the process. This model consists of:

- pyrolysis and oxidation of non-aromatic species;
- initial soot precursors formation from non-aromatic species;
- polycyclic aromatic hydrocarbon growth through the hydrogen abstraction and acetylene addition reaction mechanism;
- soot particles nucleation through coalescence of polycyclic aromatic hydrocarbons;
- surface reactions for the carbon clusters and soot particles;
- particles growth through coagulation mechanism;
- particles, carbon clusters and precursors oxidation.

The developed kinetic mechanism for modeling a high luminosity gas burner includes concentrations of 52 species, 76 reactions of soot formation, 226 combustion reactions, heat balance for the pyrolysis and combustion processes.

Unknown rate coefficients for soot formation reactions were calculated using Dr. Chemie computer code [1,2].

Results

The modeling of the preliminary direct methane heating by burning the part of methane was performed. Then the calculations of soot formation during the methane oxidation were carried out. Input data for inlet mixture composition and temperature were obtained from the results of the direct preheating modeling.

The goals for the modeling were the following:

- (i) to explore numerically the effect of the preliminary heating on the process of soot formation;
- (ii) to understand the mechanism of the soot formation in the methane-oxygen flame and determine key factors affecting the luminosity of the flame;
- (iii) to find an optimal residence time in the preheating stage for the most effective production of soot particles on the stage of combustion based on the proposed mechanism.

The considered temperatures of preheating were in the range of $900 \div 1600$ K. The main results are the following.

Dependence of the preheating temperature on the part of burned methane for the first stage.

The employed model considered the direct heating of methane. Combustion products (CO_2 , H_2O) are mixed after precombustion with all preheated gas flow. The calculations of steady state composition [3] for different mole fractions of burned methane were performed

using NASA program package for thermodynamic computations [4,5]. The input parameter for each computation was the fraction of burned methane. Then the chemical equilibrium for the given fraction of burned methane for total gas volume was computed. Formation of soot and solid carbon black was prohibited. The results of computations include all thermodynamic parameters for quasi-equilibrium steady state mixture after precombustion, including chemical composition, energy content, temperature, etc.

The key results for temperature interval 700-1600 K are presented in the Table 1

Table 1 Temperature of preheating in dependence of the quantity of burned methane.

Vol. % CH ₄ burned	Vol. % O ₂ added on preliminary stage	Temperature of preheating, K
2.4	4.7	700
3.1	6.2	800
3.8	7.6	900
4.5	9.0	1000
5.2	10.4	1100
5.9	11.8	1200
6.7	13.4	1300
7.5	14.9	1400
8.3	16.4	1500
9.1	17.9	1600

Overall mechanism of soot formation

Combustion calculations for the developed kinetic model were conducted using Sandia code [6]. This code provides numerical solution of the system of differential equations for the employed kinetic model, and outputs time dependent values of concentrations for all species, enthalpy, and gas temperature. Typical examples of computational results for methane pyrolysis and combustion with and without precursors are presented in Figs 4 - 6 correspondingly. The

temperature profile and profiles of concentration for CH_4 , CO_2 , H_2O , C_2 containing species, C_{14} , C_{16} , C_{18} containing particles, soot particles of average size $0.3 - 1 \mu$ and average number of carbon atoms 10^8 vs. time are presented. It is shown that 0.03% of pyrene additive increases the amount of generated soot more than by the order of magnitude (see Figs 5, 6, and also see Table 3).

Analysis of the computational results for second combustion stage suggests that main reaction pathway can be divided into two main canals (see reaction scheme in Fig. 1). The right branch on the chart corresponds to different mechanisms of direct oxidation of methane. Under conditions of interest, approximately 99% of methane is oxidized up to CO_2 and H_2O and do not contribute into soot formation. The left branch of the chart leads to acetylene formation, that greatly contributes into formation of soot precursors. All carbon atoms, which will not be oxidized during the evolution into acetelene, will form the soot precursors. The amount of these carbon atoms under given conditions is about 0.1%. Then soot precursors form very fast (comparing with oxidation) soot particles through the processes of acetylene addition, nucleation, coagulation. And, consequently, the oxidation of soot particles takes place.

Soot growth mechanism and role of preliminary heating

Kinetic chart of soot growth mechanism is presented in Fig. 2. The overall evolution of carbon atoms from methane to soot can be divided into stages. Stage 1 corresponds to the fast loss of carbon atoms trough the oxidation of intermediate species. The rate of oxidation exceeds significantly the rate of cluster formation. Stage 2 corresponds to the fast (compared to the rate of oxidation) carbon clusters growth.

The formation of precursors with the number of carbon atoms more than C_{16} (pyrene, see Fig. 3) has the threshold character. Before this threshold the carbon atoms are lost, mostly through the oxidation. After the threshold the carbon atoms participate in the soot formation and growth. The total numbers of carbon atoms transformed into higher C_n states during the time evolution are shown on Fig. 2. Without preheating 1% of carbon atoms is transformed into C_2H_2 , 0.1% into pyrene and almost 0.1% into soot. Preheating helps to crack methane at Stage 1 and to produce precursors. The formed precursors (the concentrations of components resulting from the preliminary heating were calculated by utilizing our kinetic model, described above) reacting with acetylene on Stage 2 save acetylene from oxidation and increase drastically the concentration of soot particles. The calculations show (see Fig. 2) that it is necessary to have 1% (0.05 vol. %) of acetylene and 0.03% (2×10^{-4} vol. %) of pyrene in optimal regime to save all acetylene. These parameters define the conditions for the preliminary heating, in particular, the residence time and temperature. The characteristic parameters for the gas burner working in optimal regime are summarized in Tables 2, 3.

Finally, as shown on Figure 2, without preheating 0.1% of carbon atoms will be transformed into the soot in the optimal regime. When utilizing direct preheating of methane on the first stage with conditions given in Table 2, then 2% of initial carbon atoms will be converted into soot. That increases the amount of soot particles by factor of 20, comparing to the process without preliminary heating.

Table 2. The optimal parameters of high luminosity burner

temperature	min residence time	max residence time
-------------	--------------------	--------------------

1300 K	10^{-2} sec	1 sec
1600 K	10^{-4} sec	10^{-2} sec

Table 3. Intermediate soot characteristics

Without precursors	60 mg/m ³ (0.1%), (3×10^{-8} Vol. %)
With ~ 0,03 % (~30 mg/m ³) pyrene	1200 mg/m ³ (2%) (6×10^{-7} Vol. %)
Soot particle average size	0,3 - 1 μ
Soot number density	10^8 - 10^9 cm ⁻³

Figure 1. Chemical reaction mechanism

Conclusions

It is clearly demonstrated that the role of preliminary heating of methane consists of partial cracking and producing of soot precursors.

The presence of soot precursors during the combustion increases drastically (by the number of 20) the amount of carbon atoms directed into soot particles through the processes of acetylene addition preventing the oxidation of acetylene.

The process of soot growth has the threshold character with respect to the size of carbon particles. The threshold corresponds to the value of C_{16} .

The parameters of high luminosity burner based on the developed principles are presented in Tables 2, 3.

Figure 2. Soot growth mechanism. (Concentrations are given in percentage from initial number of carbon atoms in CH_4)

Figure 3. PYRENE C₁₆H₁₀

Melting temperature 150° C,
Boiling temperature 399° C

Figure 4. Kinetics of methane pyrolysis, $T = 1600\text{ K}$

Figure 5. Kinetics of methane oxidation without precursors

Figure 6. Kinetics of methane oxidation with precursors

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APPENDIX B

MATHEMATICAL MODELING OF RADIATION HEAT TRANSFER IN A NOVEL, HIGH-HEAT TRANSFER, LOW NO_x NATURAL GAS COMBUSTION SYSTEM

Purdue University

EXECUTIVE SUMMARY

Analysis has been developed to predict radiation heat transfer from luminous flames in a model glass melting furnace using natural gas as a fuel and industrial oxygen as oxidizer. Parametric calculations have been performed to gain improved understanding of heat transfer needed for the design of a novel low NO_x , high luminosity burner for oxygen-fired glass melting furnaces. A systematic approach has been followed in developing more realistic models by relaxing some of the idealizing simplifications made in the "well-stirred" furnace model. As a reference for assessing the thermal performance of luminous flames, calculations are first performed for a nonluminous flame with H_2O and CO_2 as the only important combustion products.

Effects of the specific firing rate, fuel and soot burnout length, soot volume fraction, oxidizer inlet temperature, furnace geometry such as height and length as well as other design factors are studied parametrically. The results show that the soot volume fraction at the furnace inlet and fuel as well as soot burnout lengths are important model parameters which affect the furnace thermal performance. Results of calculations have indicated that for sufficiently large soot volume fractions the performance for luminous flames could be as much as 25% better than for nonluminous flames under otherwise the same operating conditions. The calculations have also revealed that flames which cover a larger fraction of the glass surface result in enhancement of the thermal performance of the furnace. Short flames are undesirable as they produce highly peaked and nonuniform heat fluxes at the glass surface, high gas and refractory temperatures.

NOMENCLATURE

c_p	specific heat at constant pressure
F	radiative flux vector
f_v	soot volume fraction
H	height of model furnace, see Figs. 2.1 and 3.1
ΔH_c	heat of combustion of fuel
h	heat transfer coefficient or enthalpy
I	intensity of radiation
k	thermal conductivity or imaginary part of complex index of refraction
L	length of model furnace (see Fig. 3.1)
m_f	fuel flow rate
m_o	oxidizer flow rate
n	real part of complex index of refraction
P	total pressure
p	partial pressure
T	temperature
U_c	overall heat transfer coefficient through crown
W	width of model furnace
α	absorptance or absorptivity
ε	emittance or emissivity
η	heat transfer efficiency defined by equation (3.6)
κ	absorption coefficient
ρ	reflectivity
σ	Stefan-Boltzmann constant
τ	transmittance
Ω	solid angle

Subscripts

amb	–	refers to ambient
b	–	refers to blackbody
c	–	refers to crown
f	–	refers to fuel
G	–	refers to glass surface
g	–	refers to gas
m	–	refers to mean beam length
r	–	refers to radiation
s	–	refers to soot
t	–	refers to total (gas plus soot)
λ	–	refers to wavelength
ω	–	refers to wavenumber
1	–	refers to glass surface (see Fig. 2.1)
2	–	refers to crown (see Fig. 2.1)

1. INTRODUCTION

1.1 Background

During the past several years, many glass manufacturers have chosen to convert one or more of their furnaces to oxy-fuel firing [1-3]. The main issues driving the conversions were NO_x and particulate emissions, fuel efficiency and checker problems. The main incentives for future conversions continue to be reduction of pollutant emissions, increase in productivity, and improvement in glass quality. The main disadvantages of oxy-gas combustion are: 1) more nonuniform temperature distribution which leads to hot spots and reduced crown refractory life, 2) non-luminous flame when fired in a hot furnace, and 3) nonuniform radiation heat flux distribution to the load in the furnace. One of the methods of overcoming these limitations of oxy-fuel fire is to develop a low NO_x burner which would improve energy efficiency by increasing heat transfer to the load. In the novel burner concept soot particles are generated by cracking natural gas. The soot particles when burned increase the flame luminosity and result in a higher heat transfer rate to the load, and at the same time lower the flame temperature. As a consequence, the heat transfer uniformity is increased and pollutant emissions are reduced.

Fundamentals of molecular gas radiation by combustion products (H_2O , CO_2 , CH_4 , CO , SO_2 , NO , C_2H_2 , N_2O and NH_3) are reasonably well in hand [4,5], and engineering methods have been developed for calculating radiation heat transfer in combustion systems [6,7]. The level of detail needed in the calculations depends greatly on the specific applications. The models required for radiation heat transfer calculations can be much more forgiving than those needed to local flame temperatures and eventually the chemical reaction rates and pollutant emissions. Hottel and Sarofim [8] have demonstrated that water vapor and carbon dioxide as the dominant radiating gases in fossil fuel combustion systems. In non-sooty natural gas flames, the

emission and absorption of radiation by these gases is usually dominant that it has become customary to neglect the contribution of other gases in the analysis of radiative heat transfer in combustion systems. In luminous flames continuous radiation from the soot particles can become significant relative to the molecular gas radiation and must be considered in calculating radiation heat transfer from such flames [9].

1.2 The Aim of this Study

Radiation heat transfer from a flame to a glass melt can be considered as consisting of two components: a part that is directly radiated to the glass, and another part is radiated to the glass via the refractory. The radiation emitted by the flame is partly reflected at the refractory and is redirected towards the glass. This is illustrated schematically in Fig. 1.1. Available results [10] show that the radiation component from the refractory along the furnace can be significant accounting from about 35 to 55% of the total heat flux. Hence, the reflectivity and emissivity of the refractory is expected to be a property which can impact the radiation heat flux to the glass in a furnace.

The radiation from nonluminous natural gas flame is mainly emitted in the infrared by the combustion gases H_2O and CO_2 . These gases are not continuous emitters but emit and absorb radiation in a number of spectral bands. Such bands consist of a very large number of spectral lines that can more or less overlap. These lines are caused by quantized energy transitions of the gas molecules. Between the bands the flame is nearly transparent and non-emitting. A typical calculated flame emission spectrum is shown in Fig. 1.2. Such a spectral emitted radiance results from the spectral emittance of the gases, multiplied by the spectral intensity of blackbody radiation.

Owing to the spectral radiation characteristics of the gas, the emissivity of the refractory is expected to influence the irradiation (incident flux) at the glass surface and the net radiative heat flux. If the refractory emissivity is low, most of the banded radiation is reflected. The wavelength distribution is unchanged by reflection and the reflected flux is strongly absorbed by the emission/absorption bands of the gas. If, on the other hand, the refractory is strongly absorbing and the emissivity is high, most of the flame radiation is absorbed and re-emitted in a more continuous wavelength distribution. The transparent wavelength regions of the flame spectrum will transmit a great part of this radiation, so that a relatively large irradiation reaches the glass surface. The conclusion is that a high emissivity refractory promotes radiation heat transfer from the flame to the glass. A compensating effect will be that the flame becomes less hot so that the direct flux from the flame to the glass is decreased. However, the total effect is still such that a strongly absorbing refractory is still expected to be favorable.

The main aims of this study are twofold:

1. Develop a simple description of radiation heat transfer from an oxygen-fired natural gas plus soot flame to the glass surface in a model glass melting furnace.
2. Perform parametric calculations using the models developed and provide guidance to IGT for developing an efficient, low NO_x natural gas combustion system for a glass melting furnace.

1.3 Scope of the Study

It is premature to simulate the combustion chamber of a glass melting furnace, because the burner system has not been designed. Also, the state-of-the-art of modeling combustion of natural gas-soot mixtures with industrial oxygen as oxidizer has not progressed to the point

where such calculations can be done routinely. Development of models/computer programs is actively being pursued [11]. Therefore, in order to be consistent with the overall goals of the IGT/DOE project, the scope of this work is much more limited and modest.

To meet the goals of the project, a step-by-step approach has been taken. The specific tasks undertaken are the following:

1. Predict the total emittance of $\text{CO}_2/\text{H}_2\text{O}$ and $\text{CO}_2/\text{H}_2\text{O}/\text{soot}$ mixtures for conditions prototypic of high-temperature glass melting furnaces.
2. Develop a simplified, one-dimensional, energy balance-based model of a glass melting furnace fired with natural gas + soot mixture and industrial oxygen as oxidizer and perform parametric calculations to predict radiation heat transfer at the glass surface.
3. Extend the model developed under Task 2 to account for radiation heat transfer along the furnace by developing a quasi-two-dimensional and perform parametric calculations of radiation heat transfer at the glass surface.

The models and results of calculations have been presented in progress reports and need not be repeated. In this report the models are summarized, some representative results are presented and the conclusions/recommendations are highlighted.

Figure 1.1 Schematic representation of radiative heat transfer in a glass furnace combustion chamber: direct flux from flame to glass Q_d and indirect flux to glass via refractory Q_r .

Figure 1.2

Calculated distribution of spectral radiance emitted $I_{e,\lambda}$ from a homogeneous mixture of combustion gases CO_2 and H_2O and N_2 versus wavelength λ . Temperature: 2000 K; partial pressure: H_2O : 0.15 atm; CO_2 : 0.08 atm; N_2 : 0.77 atm; path length: 2.8 m. The Planck black body emitted radiance is indicated by the dotted line. The calculations are based on the statistical narrow band model.

2. WELL-STIRRED FURNACE MODEL

2.1 Introduction

Even if the chemical kinetics of the oxidation reactions are not modeled in the combustion chamber and all the fuel burns out as it is done in the "stirred furnace" model, the major challenge still remains to predict the radiation from the flame and the superstructure to the load (glass surface). A number of practical engineering methods have been developed to predict radiation heat transfer from nonluminous and luminous flames and includes the following methods [6]: 1) total emittance, 2) zonal (Hottel), 3) flux and differential approximations, 4) Monte Carlo (statistical), and 5) numerical (i.e., discrete ordinates, discrete transfer) methods. Of the methods listed the total emittance (and absorptance) based on the mean beam length concept is the simplest procedure. Yet, using this approach it is possible to include detailed band information for the infrared radiating gases. The main disadvantage of the method is that it is valid for isothermal gas volume only, and its accuracy is uncertain. In spite of these shortcomings, the method is used for the preliminary parametric calculations.

In this section of the report a highly simplified glass melting furnace combustion chamber model is constructed to facilitate preliminary calculations. The model idealizations are listed, and the analysis is further simplified to express the radiation heat flux at the glass surface in terms of the radiation characteristics (i.e., total emittance and absorption) of the soot mixture. The models for calculating the radiation characteristics are reviewed, and results of parametric calculations are reported for both nonluminous and luminous flames in terms of relevant model parameters such as gas temperature, partial pressure of the absorbing gas species times pathlength product and soot volume fraction.

2.2 Model Formulation

To set up the problem of predicting radiation heat transfer from a luminous flame and to identify the relevant model parameters, consider a highly simplified "furnace" geometry: an infinite plane layer of homogeneous (i.e., isothermal and uniform composition) gas-soot mixture, confined between two infinitely large plates of uniform, known temperature, separated at a distance H (Fig. 2.1). To indicate the different geometry used in this analysis, the two surfaces are numbers 1 (glass, load) and 2 (crown). The surfaces are taken to be diffuse emitters and reflectors of radiation. The spectral absorption coefficient of the flame (gas-soot-mixture) and the radiation properties of the surfaces 1 and 2 are assumed to be known.

One can readily show that in this plane geometry and the idealized conditions considered, the net spectral radiation heat flux at surface 1 (the glass load) can be written as

$$q_{r1,\omega} = \left\{ \left[\epsilon_{1,\omega}(1 - 2\epsilon_{2,\omega} E_3(\kappa_\omega H) - 4\rho_{2,\omega} E_3^2(\kappa_\omega H)) \right] \left[E_{bg,\omega}(T_g) - E_{b2,\omega}(T_2) \right] + 2\epsilon_{1,\omega} \epsilon_{2,\omega} E_3(\kappa_\omega H) \left[E_{b2,\omega}(T_2) - E_{b1,\omega}(T_1) \right] \right\} / \left[1 - 4\rho_{1,\omega} \rho_{2,\omega} E_3(\kappa_\omega H) \right] \quad (2.1)$$

where $\epsilon_{1,\omega}$ and $\epsilon_{2,\omega}$ are spectral emissivities of the surfaces, $\rho_{1,\omega}$ and $\rho_{2,\omega}$ are the reflectivities ($\rho_{1,\omega} = 1 - \epsilon_{1,\omega}$), and κ_ω is the spectral absorption coefficient of the flame, and $E_{bg,\omega}(T_g)$, $E_{b1,\omega}(T_1)$ and $E_{b2,\omega}(T_2)$ are the black body emitted fluxes of the gas (flame) and surfaces, respectively. Further, $E_3(x)$ is the exponential integral function defined as

$$E_3(x) = \int_0^1 \mu e^{-x/\mu} d\mu = \int_1^\infty e^{-xt} dt/t^3 \quad (2.2)$$

The first term in the numerator of equation (2.1) represents the flame contribution; the second is the contribution to the flux from surface 2 (the crown), attenuated by the gas absorption.

$$2E_3(\kappa_\omega H) \approx \exp(-\kappa_\omega L_m) = \tau_\omega \quad (2.3)$$

where τ_ω is the spectral transmittance of the gas, with the mean beam length $L_m = 1.76 H$. Since the spectral transmittance of the gas τ_ω and the spectral absorptance α_ω are related by

$$\tau_\omega + \alpha_\omega = 1 \quad (2.4)$$

we also have that

$$\alpha_\omega = 1 - 2E_3(\kappa_\omega H) = 1 - \exp(-1.76 \kappa_\omega H) \quad (2.5)$$

Further, if use is made of Kirchoff's law [5] we can write that

$$\epsilon_\omega = \alpha_\omega = 1 - 2E_3(\kappa_\omega H) = 1 - \exp(-1.76 \kappa_\omega H) \quad (2.6)$$

where ϵ_ω is the spectral emittance of the gas.

Then, using equation (2.6) in equation (2.1) we obtain

$$q_{r1,\omega} = \left\{ \epsilon_{1,\omega} \left[1 - \epsilon_{2,\omega}(1 - \epsilon_\omega) - \rho_{1,\omega}(1 - \epsilon_\omega)^2 \right] \left[E_{bg,\omega}(T_g) - E_{b1,\omega}(T_1) \right] \right. \\ \left. + \epsilon_{2,\omega}(1 - \epsilon_\omega) \left[E_{1,2\omega}(T_2) - E_{b,1\omega}(T_1) \right] \right\} / \left[1 - \rho_{1,\omega}\rho_{2,\omega}(1 - \epsilon_\omega)^2 \right] \quad (2.7)$$

The denominator of equation (2.7) can be written as a series, where each power of τ_ω represents a number of reflections at a surface. Since the reflectivity of the glass surface is small ($\rho_{1,\omega} = 0.1$), the term $\rho_{1,\omega}\rho_{2,\omega}(1 - \epsilon_\omega)^2 \ll 1$ in the denominator of equation (2.7) can be approximated as unity. Evaluation of the spectral emittance of the flame ϵ_ω from the knowledge of the spectral absorption coefficient of the gases and the soot and integration of equation (2.7) over the entire spectrum allows one to write

$$q_{r1} = \int_0^\infty q_{r1,\omega} d\omega \quad (2.8)$$

If surfaces 1 and 2 are assumed to be gray (i.e., spectrally non-selective), the glass and crown temperatures are assumed to be the same ($T_1 = T_2$), integration of equation (2.7) over the entire spectrum yields

$$q_{r1} = \epsilon_1 \left\{ \left[1 - \epsilon_2(1 - \epsilon_t) - \rho_2(1 - \epsilon_t)^2 \right] E_{bg}(T_g) - \left[1 - \epsilon_2(1 - \alpha_t) - \rho_2(1 - \alpha_t)^2 \right] E_{b1}(T_1) \right\} \quad (2.9)$$

where the total emittance of the gas plus soot mixture is defined as

$$\epsilon_t = \int_0^\infty \epsilon_{t,\omega}(T_g) E_{bg,\omega}(T_g) d\omega / \int_0^\infty E_{bg,\omega}(T_g) d\omega \quad (2.10)$$

and the total absorptance of the gas plus soot mixture is defined as

$$\alpha_t = \int_0^\infty \alpha_{t,\omega}(T_g) E_{b1,\omega}(T_1) d\omega / \int_0^\infty E_{b1,\omega}(T_1) d\omega \quad (2.11)$$

Equation (2.9) reveals that the total emittance/absorptance of the homogeneous gas plus soot mixture are the only radiation characteristics that are needed to predict the radiation heat transfer rate at the glass surface.

A simple analytical basis for determining the total emittance of gases and soot has been developed [11]. The total emittance ϵ_t of the mixture can be expressed as

$$\epsilon_t = \epsilon_g + \epsilon_s - \Delta\epsilon_{gs} \quad (2.12)$$

where ϵ_g and ϵ_s are the emittances of gas and soot, respectively, and $\Delta\epsilon_{gs}$ is the correction to account for the 2.7 μm and 15 μm overlapped spectral regions of H_2O , CO_2 and soot. The total absorptance of gases and soot mixtures can be calculated in a similar manner. Models for evaluating total emittance and absorptance are briefly reviewed in the next subsection of the report.

evaluating total emittance and absorptance are briefly reviewed in the next subsection of the report.

2.3 Overview of Radiative Transfer in Flames

Predicting spectrally complicated behavior of infrared gas radiation in fossil fuel fired furnaces has been a challenging problem for thermal engineers. During the last few decades a number of approaches have been proposed to predict radiative transfer in combustion systems [6]. The most accurate model for integrating over the spectrum and calculating radiative transfer is the line-by-line approach based on a direct numerical calculation of all the lines or of all the radiating species in the gas [12]. However, two main difficulties appear in using the approach: i) the required spectral resolution must be less than the line width (i.e., 10^{-3} to 10^{-2} cm^{-1}), which corresponds to about 10^6 spectral discretization intervals to cover the entire IR spectrum of, say, CO_2 and H_2O at 2000 K, and ii) at high temperature, accurate knowledge of high number of lines, which couple high energy molecular levels appear, and an accurate knowledge of the quantum states associated to these levels is required. Recent developments in gas molecular spectroscopy have improved knowledge of radiative properties. The line-by-line approach is being systematically used for predicting radiative transfer in low temperature atmospheric physics; however, in high temperature engineering applications approximate models characterized by spectral resolution of about 25 to 400 cm^{-1} are generally adequate, and the line-by-line results are considered only as an essential benchmark. Line-by-line calculation has been reported for $\text{H}_2\text{O} - \text{CO}_2 - \text{CO}$ -transparent gas mixtures and for temperatures up to 2000 K; however, the use of this method is impractical for most engineering applications.

The radiation property models used for engineering radiant heat transfer predictions can in general be classified into the following main types: 1) total gas emittance correlations, 2) sum-

of-the-gray gas models (SGGM), 3) exponential wide band models (EWBM), 4) narrow band models (NBM) and 5) line-by-line models (LBLM). It is beyond the scope of this discussion to describe the methods of calculating the total emittance and absorptance of homogeneous infrared radiating gas mixtures, and reference is made to a recent account [12] for a discussion of the methodologies. Suffice it to mention that the SGGM and EWBM approaches are now being used routinely in radiation heat transfer predictions and are employed in the present calculations.

There exist a number of $\text{H}_2\text{O}/\text{CO}_2$ total emittance correlations for modeling radiation heat transfer from non-luminous flames and combustion systems. Lallemand and Weber [13] have recently evaluated seven such correlations for their accuracy and range of applicability. They found that the SGG model of Smith et al. [14] was the most accurate of all the models tested. The only limitations of the model were at high pressure (> 20 atm), which is not of interest in the current application. Therefore, the Smith et al. total emittance/absorptance model for $\text{H}_2\text{O}/\text{CO}_2$ mixtures is used.

The total emittance of a homogeneous soot cloud is calculated from

$$\epsilon_s = \int_0^\infty \epsilon_{s\lambda} E_{b\lambda}(T) d\lambda / \int_0^\infty E_{b\lambda}(T) d\lambda \quad (2.13)$$

where the spectral emittance of soot is given by

$$\epsilon_{s\lambda} = 1 - \exp(-\kappa_{s\lambda} L_m) \quad (2.14)$$

with L_m being the mean beam length [8]. The spectral absorption coefficient of soot, $\kappa_{s\lambda}$, is based on the small particle limit of the Mie theory (Rayleigh theory) and is given by [5, pp. 430-431]

$$\kappa_{s\lambda} = \frac{36 \pi n_\lambda k_\lambda}{[(n_\lambda^2 - k_\lambda^2 + 2)^2 + (2n_\lambda k_\lambda)^2]} \left[\frac{f_v}{\lambda} \right] \quad (2.15)$$

where n_λ and k_λ are the real and imaginary parts of the complex index of refraction, respectively, and f_v is the soot volume fraction. The optical constants of soot (i.e., n_λ and k_λ) were taken from the published literature sources [5,9].

2.4 Results and Discussion

2.4.1 Total Emittance of H_2O/CO_2 Mixtures

Recent evaluation by Lallemand and Weber [15] of the SWGG emittance models for nonluminous flames has revealed that the Smith et al. [14] and the Copalle and Vervish [16] models yield the smallest difference between the EWB model considered as a benchmark. Therefore, the results of calculations based on the two models are compared to each other. The baseline results are then compared based on a much more detailed SNB model.

Figure 2.2 illustrates the gas emittance of a nonluminous gas as a function of the gas temperature with the partial pressure of the absorbing species and mean beam length product as a parameter. The results are for stoichiometric methane-oxygen combustion products for which $p_{H_2O}/p_{CO_2} = 2$, and the total pressure is at one atmosphere. The results above gas temperature of 2400 K are extrapolated, because experimental data were not available to Smith et al. [14] when developing the empirical gas emittance correlation. The reason that the gas temperatures were extrapolated beyond 2400 K is that oxy flames may exceed 2500 K. The results show the expected trend. For a given pL the gas emittance decreases with the gas temperature, and for a given temperature the gas emittance increases with pL.

A comparison of the total gas emittance correlations developed by Smith et al. [14] and by Copalle and Vervish [16] is given in Fig. 2.3. As the agreement is reasonably good and is within the experimental uncertainty of the wide band parameters in the original studies.

model predicts higher ϵ_g , and the discrepancy increases with temperature and pressure-mean beamlength product (Fig. 2.5). According to the results, at temperatures of 2000 K and greater the SGG model can significantly underpredict the total gas emittance in comparison to that calculated by the SNB model. The two models are based on different infrared radiation molecular parameter data, but the model predictions have not been compared against experimental gas emittance data. For the calculations to be presented in the rest of the report we use the Smith et al. [14] empirical correlation to predict the total gas emittance of $\text{H}_2\text{O}/\text{CO}_2$ mixtures.

2.4.2 Total Emittance of Soot

The dependence of the total emittance of soot on the pathlength L and on the soot volume fraction f_v are illustrated in Figs. 2.6 and 2.7, respectively. For sufficiently large pathlengths and/or soot volume fractions the soot emittance approaches unity indicating that the soot cloud radiates as a blackbody.

The available spectral complex index of refraction data of soot [5,9] reveals that there is significant difference in the dependence of the real and imaginary parts of the complex index of refraction data with wavelength. The results reported are based on the small particle limit of the Mie theory, and the absorption coefficient is expected to increase if the particles agglomerate and the soot cloud is not monodisperse but polydisperse. No correlations for mean absorption coefficients have been reported for polydispersions of soot particles not obeying the Rayleigh limit. The results reported should be taken with caution until validated against experimental data or substantiated by more detailed calculations.

2.4.3 Total Emittances of H_2O/CO_2 /Soot

The effect of soot radiation on the luminous flame radiation is presented in Fig. 2.9 as a ratio of total gas plus soot emittance to the total gas emittance vs. the partial pressure of radiating gas species times pathlength product. Results included in the figure are for a range of soot volume fractions f_v considered to be in the range of interest. Additional results are included in Progress Report No. 1 (April 18, 1996). As anticipated, the results show that soot impacts significantly the total gas plus soot emittance. The results also show that at sufficiently large partial pressure of radiating species and pathlength products the ratio becomes independent of the soot volume fraction. For larger fractions f_v (not shown) the emittance ratio $\epsilon_{total}/\epsilon_g$ is greater.

2.5 Summary

An analysis has been presented to predict radiation heat transfer in a glass melting furnace using a stirred furnace model. Under certain idealizing assumption calculation of the radiation heat flux at the glass surface reduces to determining the total gas plus soot emittance and absorptance of the luminous flame.

The results reported show that the total luminous flame emittance is significantly higher than the total gas emittance for all temperatures and partial pressure of the radiating gas species times pathlength products. The emittance results indicate that the radiation heat transfer from the luminous flame have a potential of significantly enhancing the radiation heat transfer rate to the glass surface. The amount of enhancement depends strongly on the soot volume fraction and the mean partial volume of the furnace occupied by the soot.

Figure 2.1 Schematic of the well-stirred glass melting furnace model.

Figure 2.2 Dependence of the total gas emittance on temperature as partial pressure of absorbing species times pathlength product based on Smith et al. [14] model: $p_{H_2O}/p_{CO_2} = 2$ and $P = 1$ atm.

Figure 2.3 Comparison of total gas emittance predicted by the Smith et al. [14] and Coppalle and Vervish [16] models: $p_{H_2O}/p_{CO_2} = 2$ and $P = 1$ atm.

Figure 2.4 Comparison of total gas emittances predicted by the Smith et al. [14] and the statistical narrow band [17] model: $p_{\text{H}_2\text{O}}/p_{\text{CO}_2} = 2$ and $P = 1 \text{ atm}$.

Figure 2.5 Relative error between the SGG [14] and SNB [17] models: $p_{\text{H}_2\text{O}}/p_{\text{CO}_2} \approx 2$ and $P = 1 \text{ atm}$.

Figure 2.6 Dependence of the total emittance of soot on the pathlength with temperature as a parameter.

Figure 2.7 Dependence of the total emittance of soot on the soot volume fraction with temperature as a parameter.

Figure 2.8 Dependence of the ratio of the total (gas + soot) emittance to the gas emittance on partial pressure times pathlength product with soot volume fraction and temperature as a parameter: $p_{\text{H}_2\text{O}}/p_{\text{CO}_2} = 2$ and $P = 1$ atm.

3. PLUG FLOW FURNACE MODEL

3.1 Introduction

A comparison of the total radiation characteristics of $\text{H}_2\text{O}/\text{CO}_2$ and $\text{H}_2\text{O}/\text{CO}_2/\text{soot}$ mixtures has revealed that soot has a potential of significantly enhancing the radiation heat transfer from the flame to the glass surface. However, a large number of idealizing simplifications have been made in the analysis. Probably the most serious assumption is that the combustion products are isothermal and of uniform composition. This is a major idealization of the physical reality and a limitation of the results reported. Detailed calculations of turbulent flow and mixing, chemical reactions, and heat transfer from the combustion products to the load are beyond the extent of this scoping study.

This section of the report describes the development of an idealized one-dimensional ("plug flow") model for predicting radiation heat transfer from non-luminous and luminous combustion products to the load at the bottom of the furnace. The model is intended to relax some of the idealizations discussed in the preceding section of this report. Results of parametric calculations are reported for assessing the increase in radiation heat transfer to the load due to the presence of soot in the combustion products. As a baseline condition, results are first reported for non-luminous combustion products. Then, sensitivity calculations are performed to examine the effects of the most important furnace model parameters to determine their effect on the radiation heat transfer along the load.

3.2. Model Description

In order to enable more realistic radiation heat transfer predictions to the load in a high-temperature (i.e., glass melting furnace, see Fig. 3.1 for sketch) a one-dimensional, energy

equation-based, furnace model has been developed. The idealized geometry (uniform height between crown and load surface) is considered to represent a single port of a cross-fired, regenerative glass melting furnace. The furnace is assumed to extend indefinitely in the direction normal to the paper, and there is no gas or radiation transport in this direction.

The model assumptions are the following:

- 1) The plug flow is steady and heat transfer to the load (i.e., glass surface) is one-dimensional.
- 2) Turbulence, mixing and detailed chemical reactions are neglected, and fuel burnout is prescribed by an empirical function.
- 3) Heat transfer from the fuel-combustion products to the load is by radiation and convection, and heat losses by radiation through the entrance and exit "ports" is neglected.
- 4) Radiative transfer is one-dimensional (i.e., in the direction perpendicular to the glass surface), and the radiation flux is based on the local but cross-section average gas temperature and radiating species concentrations. Radiation heat transfer from the infrared radiating gas is prescribed using a sum-of-the-gray gases model of Smith et al. [14].
- 5) The emission and absorption of radiation by soot particles is handled through the specification of the soot volume fraction [5,6].
- 6) The crown is assumed to be a gray, diffuse emitter and absorber of radiation, and interreflection of radiation between the crown and load surface is accounted for.
- 7) The convective heat transfer coefficient at the load and crown are calculated using an empirical correlation published in the literature [18]. The average heat transfer coefficient $\bar{h}$ is given by the equation,

$$Nu = \bar{h}L/k = CRe^{3/4}$$

where L is the characteristic length, k is the thermal conductivity of the gas and Re is the Reynolds number. In this equation the coefficient C for end and side firing are 0.175 and 0.142, respectively.

- 8) Heat losses through the crown are calculated by prescribing the overall heat loss coefficient between the crown surface and the ambient surroundings.
- 9) Glass is idealized to be opaque to thermal radiation of all wavelengths, and the glass surface temperature is assumed to be uniform along the furnace.
- 10) The chemical energy content of the unburned soot introduced into the furnace is negligible in comparison to that of the fuel.

Since turbulent flow and combustion reactions are not predicted, the analysis is based on the energy equation only. Employing the idealizations listed in the preceding paragraph, the thermal energy equation can be written as

$$(m_o + m_f) \frac{dh_g}{dx} = -W(q_G + q_c) + m_f \Delta H_c f(x) \quad (3.1)$$

where $f(x)$ is the fuel burn rate per unit length and the total (radiative plus convective) heat flux at the glass surface q_G is given by

$$q_G = \epsilon_G \left[(\tilde{\epsilon}_g \sigma T_g^4 - \tilde{\alpha}_g \sigma T_G^4) + \epsilon_c \tau_g (\sigma T_c^4 - \sigma T_G^4) \right] + \bar{h}_G (T_g - T_G) \quad (3.2)$$

and the heat flux at the crown q_c is given by

$$q_c = \epsilon_c \left[(\tilde{\epsilon}_g \sigma T_g^4 - \tilde{\alpha}_g \sigma T_c^4) + \epsilon_G \tau_g (\sigma T_G^4 - \sigma T_c^4) \right] + \bar{h}_c (T_g - T_c) \quad (3.3)$$

Once the enthalpy of the gas h_g has been calculated, the gas temperature T_g can be determined

from the knowledge of the scientific heat of the gas such that

$$h_g = \int_{T_{g0}}^{T_g} c_p(T) dT \quad (3.4)$$

where T_{g0} is the reference temperature. The local crown temperature T_c is calculated from an energy balance at the crown/gas interface

$$q_c(x) = U_c(T_c - T_{amb}) \quad (3.5)$$

where U_c is the overall heat transfer coefficient of the crown, including convective and radiative heat transfer from the superstructure of the furnace to the ambient surroundings.

The gas (oxygen plus fuel) temperature at the inlet was prescribed, and the nonlinear system of the model equations, equations (3.1)-(3.5), is solved numerically using a finite-difference method which required iterative solution. Grid independence study was conducted before generating the numerical results of the parametric study. For a 10 m long model furnace 50, 100 and 200 nodes were employed. The results for 100 and 200 nodes differed very little; therefore, the calculations were performed using 100 nodes. An energy balance on the furnace was made and the balance was closed within 0.1% of the fuel input rate.

3.3 Results and Discussion

3.3.1 Scope of Parametric Studies

Even for a very simple plug flow model developed and used here, there are a large number of parameters controlling heat transfer from the combustion products to the load. The parameters include the following:

1. Furnace design and materials (see Fig. 3.1)

- Height H

- Length L
- Radiative properties of refractories
- Crown refractories and their thicknesses

2. Furnace operating conditions

- Specific firing rate, SFR
- Oxidizer preheating temperature
- Fuel burnout length
- Load (glass) surface temperature
- Stoichiometry

3. Soot conditions

- Soot volume fraction at inlet to furnace f_v
- Soot burnout length.

An attempt has been made to identify the most important model parameters, and systematic parametric calculations were then performed to determine the effect of these parameters on the thermal performance of the combustion system. Before presenting the results of parametric calculations a baseline simulation has been performed and the results are discussed to obtain feel for the thermal performance of the system. The results obtained should not be considered as absolute, but should be taken as relative.

3.3.2 Baseline Simulation

The model parameters for the furnace design and operating conditions chosen were

somewhat arbitrary but are considered to be representative of a glass melting furnaces. They were as follows:

Height, $H = 1.5$ m

Length, $L = 10.0$ m

Crown emittance, $\epsilon_c = 0.6$

Overall crown heat transfer coefficient, $U_c = 20$ W/m²K

Glass surface temperature, $T_G = 1800$ K

Specific fuel firing rate, $SFR = 2.0$ MW/m²

Stoichiometric combustion

Preheated oxygen at inlet to furnace, 1500 K

Soot volume fraction at inlet to chamber, $f_v = 0$.

For reference, the fuel burnout length L_b is defined as a length by which 99% of fuel has burned out.

The variation of the combustion products (gas) and crown temperature along the model furnace is illustrated in Fig. 3.2. The maximum gas temperature occurs at about $x/L = 0.18$. At this location the gas to crown temperature difference is quite large (~ 500 K), but decreases rapidly and becomes almost constant at the middle of the furnace. This suggests that the furnace is sufficiently long to transfer the sensible energy from the combustion products to the load.

The variation of the local heat flux along the furnace is illustrated in Fig. 3.3. A comparison of the gas temperature in Fig. 3.2 and the total heat flux occur at the same location along the furnace as expected based on the model idealizations. The average convective heat flux is less than one percent of the radiative heat flux and can be neglected. For $x/L < 0.05$ the local heat flux is negative because heat is extracted from the gas when the assumed glass surface

temperature is high (= 1800 K). The strong peak in heat flux is undesirable, as this may cause glass reboiling and shorten crown refractory lifetime [2]. The heat transfer effectiveness (efficiency) of this model furnace defined as

$$\eta = \frac{\text{Heat transfer rate to glass}}{\text{Fuel firing rate}} = \frac{\int_0^L q(x)W dx}{\text{SFR} \times H \times W} = \frac{\bar{q} \times L}{\text{SFR} \times H} \quad (3.6)$$

where the width of the furnace W is taken to be 1 m. With the heat losses only through the crown, and absence of losses through the sides (ends), the heat transfer effectiveness calculated is 65%, which is much higher than for regenerative, side-fired glass melting furnaces in industrial practice. This effectiveness is for stoichiometric oxygen to fuel ratio and preheated oxygen air temperature of 1500 K at the inlet to the furnace. Hence, because of the idealizing approximations made in the analysis, the results should be interpreted with caution and be considered as relative.

3.3.3 Effect of Specific Firing Rate

All of the furnace model parameters were kept the same as in the baseline simulation, except the specific firing rate (SFR) was varied parametrically. The effect of SFR on the combustion products temperature, heat flux and gas emittance distributions along the furnace are illustrated in Figs. 3.4, 3.5 and 3.6, respectively. The trends are as expected. With increasing specific firing rate both the local gas (Fig. 3.4) and the local heat flux (Fig. 3.5) distributions increase, reach their peaks and then decrease rather sharply. The gas temperature decreases sharply as the thermal energy is radiated. Since the emittance of H_2O and CO_2 decrease with temperature, the trends predicted in the gas emittance (Fig. 3.6) are consistent with the variation in the gas temperature shown in Fig. 3.4.

The thermal performance results are summarized in Table 3.1. The maximum gas and crown temperatures, the average heat flux and the heat transfer effectiveness increase with the specific firing rate SFR. The maximum gas temperatures and effectiveness at $\text{SFR} = 4 \text{ MW/m}^2$ appear to be unreasonably high and suggest the failure of the highly idealized parametric model to predict the thermal performance of the model furnace. The two main factors that contribute to this is the neglect of dissociation of the combustion products at high temperatures and of radiative transfer in the axial (flow) direction by the products. It is well known that at temperatures of about 2200 K water vapor begins to dissociate significantly, and the energy required to dissociate the molecule would decrease the gas temperature as a larger fraction of molecules dissociate with increasing gas temperature. Also, at high temperatures nonequilibrium effects will have to be considered in order to make realistic predictions of chemical kinetics and the rate of combustion.

Table 3.1 Effect of specific firing rate (SFR) on the thermal performance on the model furnace

SFR (MW/m ²)	T _{g,max} (K)	T _{c,max} (K)	$\bar{q}$ (W/m ²)	η (%)
1	2314	1978	0.0833	55.5
2	2625	2114	0.195	65.0
4	3010	2282	0.410	75.8

3.4 Effect of Fuel Burnout Length

The fuel burnout length is an important model parameter. It controls the rate of conversion of chemical to thermal energy and hence impacts greatly the local gas temperature and the heat flux to the load. The longer the flame the more gradual is the heat release. Hence, the more uniform are the gas and crown temperatures and the heat flux to the load [2]. The fuel

burnout length was varied parametrically (see Fig. 3.7) from the baseline simulation, and the results for the gas temperatures, local heat fluxes to the load and gas emittances are illustrated in Figs. 3.8, 3.9 and 3.10, respectively. The results show that as the relative burnout length L_b/L increases, the peak gas temperatures and heat fluxes decrease and become more uniform. This is a desirable trend and is well recognized in practice [2]. The variation of the local gas emittance along the furnace with the relative fuel burnout length L_b/L (Fig. 3.10) reveals the expected behavior of the emittance with the gas temperature [6].

The thermal performance results are summarized in Table 3.2. The results included in the table reveal that the greatest impact of L_b/L is on the maximum gas temperature. The average heat flux to the load and the heat transfer effectiveness are increased only by a few percent after increasing the relative fuel burnup length L_b/L from 0.2 to 0.6.

Table 3.2 Effect of relative fuel burnup length L_b/L on the thermal performance of the model furnace

L_b/L (-)	$T_{g,max}$ (K)	$T_{c,max}$ (K)	$\bar{q}$ (MW/m ²)	η (%)
0.2	3035	2293	0.187	62.4
0.4	2625	2114	0.195	65.0
0.6	2429	2028	0.200	66.5

3.3.5 Effects of Other Model Parameters

The effects of glass surface temperature, oxygen preheating, furnace height and length, crown emissivity and crown overall heat transfer coefficient have been studied. Detailed results are included in Progress Report No. 2 (July 31, 1996). In summary, the crown emissivity and crown overall heat transfer coefficient were found to be unimportant model parameters for the furnace geometry and operating conditions considered. Glass surface temperature, oxygen

preheating, furnace height and furnace length were found to be more important. They affected the heat transfer rate to the glass surface and furnace efficiency. For example, a shorter furnace with a fixed fuel burnout length ($L_b/L = 0.4$) yielded better performance. This is owing to the fact that with a longer furnace, the heat losses to the ambient surroundings increase, the gas temperature decreases and thermal efficiency is also reduced. Increasing the oxygen temperature from 300 K to 1500 K at the inlet to the thermal efficiency from 46.3 to 65.0 percent.

3.3.6 Effect of Soot Volume Fraction

In the simulations discussed up to now the fuel introduced into the furnace did not contain any soot (i.e., soot volume fraction at the inlet to the combustion chamber $f_{v,o}$ was assumed to be zero). To examine the effect of soot on radiation heat transfer and thermal performance of the furnace, the soot volume fraction was varied parametrically. In the simulations performed, the soot volume fraction at the inlet to the chamber $f_{v,o}$ was specified and then allowed to burn out before exhausting the combustion products to the atmosphere.

The effect of $f_{v,o}$ on the local gas temperature distribution, heat flux and total (gas plus soot) emittance variation along the model furnace are illustrated in Figs. 3.11, 3.12 and 3.13, respectively. The results show that as more soot is added to the reactant stream, the total (gas plus soot) emittance of the combustion products (Fig. 3.13) and the heat flux to the load (Fig. 3.12) increases, and this decreases the gas temperature (Fig. 3.11); however, the crown temperature (not shown) increases. Inspection of Fig. 3.11 shows that the incremental decrease in the gas temperature with an increase in soot volume fraction from $f_{v,o} = 5.0 \times 10^{-7}$ to 10×10^{-7} is rather small. This suggests that the combustion products are nearly "saturated" with soot. This is supported by the results of Fig. 3.13 which shows that for $f_{v,o} \geq 5 \times 10^{-7}$ the total gas emittance is very close to unity. The emittance increases very sharply with $f_{v,o}$ particularly

near the inlet of the furnace. The results of Fig. 3.14 show that the emittance ratio (i.e., the total gas plus soot emittance to the gas emittance) reaches a value of about 3 near to $x/L = 0.2$ at $f_{v,0} = 1 \times 10^{-6}$ and approaches unity at the point where all the fuel and soot burn out ($x/L \approx 0.4$).

Table 3.3 Effect of soot volume fraction at inlet on the thermal performance of the model furnace for $L_b/L = 0.4$.

$f_{v,0}$ (-)	$T_{g,max}$ (K)	$T_{c,max}$ (K)	$\bar{q}$ (MW/m ²)	η (%)	$\bar{q}/\bar{q}_0$ (-)
0	2625	2114	0.195	65.0	1.00
1×10^{-7}	2461	2254	0.233	77.6	1.20
2.5×10^{-7}	2421	2335	0.242	80.8	1.24
5.0×10^{-7}	2408	2371	0.244	81.3	1.25
10.0×10^{-7}	2403	2377	0.243	81.1	1.25

The effect of the inlet soot volume fraction on the thermal performance of the model furnace are summarized in Table 3.3. Examination of the table reveals that the maximum crown temperature T_{max} , the average heat flux and the heat transfer effectiveness all increase with an increasing soot volume fraction $f_{v,0}$, but, as expected, the maximum gas temperature decreases with $f_{v,0}$. The last column of Table 3.3 shows that for $f_{v,0} = 1 \times 10^{-6}$, the average heat flux to the load has been increased by 25 percent compared to the average heat flux $\bar{q}_0$ in the absence of soot ($f_{v,0} = 0$) in the reactant stream. The potential improvement in the performance of the furnace is the more surprising is that in the process the maximum gas temperature has been reduced by more than 200 K. As a consequence of the reduction in the gas temperature, the NO_x emissions are expected to decrease. The only deleterious effect of the soot in the reactants is the increase in the crown temperature which would impact the lifetime of the superstructure.

3.3.7 Effect of Soot Burnout Length

The initial soot volume fraction $f_{v,o}$ was held constant at 0.25×10^{-7} . The relative fuel burnout length was also held constant at $L_b/L = 0.4$, but the relative soot burnout length $L_{b,s}/L$ was varied parametrically to determine its effect on the thermal performance of the combustion chamber. The effects of $L_{b,s}/L$ on the local gas temperature, heat flux to the load and the total combustion products emittance variation along the model furnace are illustrated in Figs. 3.15, 3.16 and 3.17, respectively. Increase in $L_{b,s}/L$ results in an increase in the average heat flux, heat transfer effectiveness and produces a decrease of the maximum gas temperature. The increase in performance is relatively modest ($< 15\%$) but with other modifications in the combustion system design and operating conditions, the potential improvements in the system performance could be significant. An increase in $L_{b,s}/L$ increases the total combustion products emittance in the region where the soot burns out but not beyond (Fig. 3.18).

Table 3.4 Effect of soot burnout length on the thermal performance of the model furnace for $f_{v,o} = 2.5 \times 10^{-7}$ and $L_b/L = 0.4$

$L_{b,s}/L$ (-)	$T_{g,max}$ (K)	$T_{c,max}$ (K)	$\bar{q}$ (MW/m ²)	η (%)	$\bar{q}/\bar{q}_0$ (-)
0.2	2521	2207	0.216	71.9	1.11
0.4	2421	2335	0.242	80.8	1.24
0.6	2415	2359	0.248	82.8	1.28
0.8	2413	2365	0.249	83.1	1.28

The effects of $L_{b,s}/L$ on the thermal performance of the combustion chamber are summarized in Table 3.4. In spite of the decrease in the combustion products temperature with an increase in $L_{b,s}/L$, the thermal performance of the chamber increases. The results contained in the figures (Fig. 3.15, 3.16 and 3.17) suggest that the thermal performance can be further improved by staging the introduction of sooty fuel mixture and of the oxidizer into the

combustion chamber. There appears to be a potential of optimizing the thermal performance of the combustion system by appropriate burner-furnace design and operating conditions. Preliminary calculations have shown that the heat transfer effectiveness can be substantially improved and the maximum gas temperature decreased by such a firing scheme.

3.3.8 Comparison of Furnace Performance for Oxygen and Air as Oxidizers

To put the model predictions in perspective, a single simulation has been performed with air as the oxidizer under stoichiometric conditions. All the other model parameters were the same as for the baseline simulation. A comparison of the gas temperatures with oxygen and air as oxidizers is given in Fig. 3.19 and of local heat flux distribution at the load surface in Fig. 3.20. The results show that both the peak gas temperature and heat flux are reduced with fuel/air firing in comparison to that with fuel/oxygen firing. This is consistent with findings in practice.

Table 3.5 Comparison of thermal performance results for O_2 and air firing the model furnace with $L_b/L = 0.4$, $f_{v,o} = 0$ and $T_{ox} = 1500$ K.

Oxidizer (-)	$T_{g,max}$ (K)	$T_{c,max}$ (K)	$\bar{q}$ (MW/m ²)	η (%)
O_2	2625	2114	0.195	65
Air	2295	1969	0.155	51

Table 3.5 summarizes the thermal performance results for the two oxidizers. The table reveals that the heat transfer effectiveness and the maximum gas temperatures are significantly higher for O_2 -firing. The trends in the results are consistent with observations in glass manufacture.

3.4 Summary

A simplified, one-dimensional {"plug flow"} combustion chamber model has been

developed for luminous radiation heat transfer in a high-temperature glass melting furnace.

Based on the extensive parametric calculations the following conclusions can be drawn:

- 1) In the absence of soot in the fuel at the inlet to the combustion system, the local gas temperature and heat flux to the load are strongly peaked.
- 2) In the absence of soot in the fuel and combustion products, the relative fuel burnout length and the oxidizer temperature are important model input parameters which influence the combustion performance, whereas the crown emittance and the overall crown heat loss coefficients are of secondary importance.
- 3) The soot volume fraction at the inlet to the combustion chamber can have a very significant impact on the furnace thermal performance. The results show that if the soot volume fraction at the inlet to the furnace of 1×10^{-6} can be achieved, the heat transfer effectiveness can be increased by at least 25 percent.
- 4) The relative soot burnout length is an important factor impacting significantly the heat transfer effectiveness to the load. The results suggest that staging of combustion and/or soot introduction into the chamber may be highly desirable for improving the heat transfer effectiveness of the combustion system.

The limitations and inadequacies of the highly simplified, one-dimensional glass melting furnace model are well recognized, but the scope of the work in this phase of the project did not warrant a detailed three-dimensional burner-furnace model for several reasons, including lack of understanding of chemistry and physics of soot burnout in a high temperature furnace and the mathematical-numerical complexity in solving model equations for turbulent flow, chemical kinetics, heat and radiative transfer.

Figure 3.1 Schematic of the plug flow, one-dimensional furnace model.

Figure 3.2 Variation of gas and crown temperatures along the model furnace for the baseline simulation.

Figure 3.3 Local heat flux distribution along the model furnace for the baseline simulation.

Figure 3.4 Effect of the specific firing rate on the gas temperature along the furnace.

Figure 3.5 Effect of specific firing rate on the heat flux distribution along the model furnace.

Figure 3.6 Effect of specific firing rate on the gas emittance distribution along the model furnace.

Figure 3.7 Variation of the fuel burnup factor α on the relative fuel burnup length L_b/L .

Figure 3.8 Effect of relative fuel burnup rate L_b/L on the gas temperature along the model furnace.

Figure 3.9 Effect of relative fuel burnup length L_b/L on the heat flux distribution along the model furnace.

Figure 3.10 Effect of the relative fuel burnup length L_o/L on the gas emittance along the model furnace.

Figure 3.11 Effect of soot volume fraction on the gas temperature along the model furnace.

Figure 3.12 Effect of soot volume fraction on the local heat flux to the load along the model furnace.

Figure 3.13 Effect of soot volume fraction on the local gas emittance variation along the model furnace.

Figure 3.14 Effect of soot volume fraction on the ratio of the total (gas + soot) emittance to the gas emittance variation along the model furnace.

Figure 3.15 Effect of relative soot burnout length $L_{b,s}/L$ on the gas temperature of a model furnace for $f_{v,o} = 2.5 \times 10^{-7}$ and $L_v/L = 0.4$

Figure 3.16 Effect of relative soot burnout length $L_{b,s}/L$ on the local heat flux distribution along the model furnace for $f_{v0} = 2.5 \times 10^{-7}$ and $L_b/L = 0.4$.

Figure 3.17 Effect of the relative soot burnout length $L_{b,s}/L$ on the total gas emittance along the model furnace for $f_{v,o} = 2.5 \times 10^{-7}$ and $L_b/L = 0.4$.

Figure 3.18 Effect of the relative soot burnout length $L_{b,s}/L$ on the local to gas emittance ratio variation along the model furnace for $f_{v,o} = 2.5 \times 10^{-7}$ and $L_b/L = 0.4$.

Figure 3.19 Comparison of gas temperatures along the model furnace for O_2 and air firing with $L_b/L = 0.4$, $f_{v,o} = 0$ and $T_{ox} = 1500$ K.

Figure 3.20 Comparison of local heat fluxes along the model furnace O₂ and air firing with $L_b/L = 0.4$, $f_{vo} = 0$ and $T_{ox} = 1500$ K.

4. QUASI-TWO-DIMENSIONAL FURNACE MODEL

4.1 Introduction

In the IGT novel burner concept intended to reduce NO_x emissions and improve energy efficiency, the soot particles are generated by cracking natural gas. The soot particles when burned increase the flame luminosity and result in a higher heat transfer rate to the load (i.e., glass or batch surface in a glass melting furnace). Predictions of radiation heat transfer from $\text{H}_2\text{O}/\text{CO}_2$ /soot in the flame and in the products to the glass load maintained at a uniform temperature have been made using a one-dimensional "plug flow" furnace model which was described in the previous section.

Both the heat transfer efficiency η and the heat transfer ratio $\bar{q}/\bar{q}_0$ were found to increase by about 25% as the soot volume fraction at the inlet to the furnace increases from 0 to 1×10^{-6} . One of the more serious shortcomings of the one-dimensional (1-D) furnace model is that it does not account for the "long-range" radiative transfer in the flow (axial) direction. Since the gas temperature is strongly peaked in the flame region and the gas-soot opacities are in the self-absorbing range, radiative transfer in the x-direction (see Fig. 3.1) is expected to be significant in comparison to the transverse direction. As a consequence, this transport is expected to decrease the peak gas temperature and make it more uniform. In turn, the nonuniformity in the heat flux distribution along the load will also be decreased. A decrease in the maximum gas and crown temperatures as well as enhancement in heat flux uniformity along the load would be highly desirable from the furnace operation point of view.

In order to relax the idealization made in the "plug flow" model (Section 3) and to examine the premise made, a quasi-two-dimensional (Quasi-2-D) model has been developed to assess the importance of the axial radiative transfer on the thermal performance of the furnace.

A two-dimensional radiative transfer model is developed to predict heat transfer in the model furnace, and this model is then coupled to the thermal energy model of the furnace to calculate the temperature distribution and heat flux along the load. Parametric calculations are then performed to examine the effects of the most important furnace design and operating parameters on its thermal performance.

4.2. Analysis

4.2.1 Model Description

In order to enable more realistic predictions of radiation heat transfer to the load in a high-temperature (i.e., glass melting furnace, see Fig. 4.1 for sketch) a quasi-two-dimensional, energy equation-based, furnace model has been developed. The idealized geometry (uniform height between crown and load surface) is considered to represent a single port of a cross-fired, regenerative glass melting furnace. The furnace is assumed to extend indefinitely in the direction normal to the paper, and there is no gas or radiation transport in this direction.

The model assumptions are the following:

- 1) The flow and heat transfer are steady, and turbulence, mixing and detailed chemical reactions are neglected. Fuel burnout is prescribed by an empirical function.
- 2) Heat transfer from the fuel-combustion products to the load is by radiation and convection, and heat losses by radiation through the entrance and exit "ports" is accounted for.
- 3) Radiative transfer is two-dimensional (i.e., in the direction perpendicular and parallel to the glass surface), and the radiation flux is based on the local gas temperature and radiating species concentrations. The radiation heat transfer model is described in greater detail in the next subsection.

- 4) The crown is assumed to be a gray, diffuse emitter and absorber of radiation, and interreflection of radiation between the crown and load surface is accounted for. Heat losses through the crown are calculated by prescribing the overall heat loss coefficient between the crown surface and the ambient surroundings.
- 5) The convective heat transfer coefficient at the load and crown are calculated using an empirical correlation published in the literature [18].
- 6) The load surface temperature is assumed to be uniform along the furnace.
- 7) The chemical energy content of the unburned soot introduced into the furnace is negligible in comparison to that of the fuel.

Based on the above idealizations, the thermal energy equation is

$$(m_o + m_f) \frac{\partial h_g}{\partial x} = -WHV \cdot F + m_f \Delta H_c f(x) \quad (4.1)$$

where m_o and m_f are the mass flow rates of oxidizer and fuel, respectively, h_g is the gas enthalpy, H and W are the height and width of furnace, respectively, ΔH_c is the heat of combustion, and $f(x)$ is an empirical function which describes the fuel burnout per unit length. The model for the radiative flux vector divergence $\nabla \cdot F$ is described in the next subsection. At the inlet to the furnace, $x=0$, the temperature of the fuel and oxidizer are prescribed. The calculation of the enthalpy (temperature) distribution is advanced by solving the thermal energy equation (2) along the furnace.

4.2.2 Radiative Transfer Model

The divergence of the radiative heat flux vector $\nabla \cdot F$ required in the thermal energy equation (1) can be obtained from the radiant energy equation. Assuming that scattering by soot

is negligible in comparison to absorption, such an energy equation for a gas-soot mixture can be written as

$$-\nabla \cdot \mathbf{F} = \int_0^\infty \kappa_\lambda \left[\int_{\Omega=4\pi} I_\lambda(\mathbf{r}, \Omega) d\Omega - 4\pi I_{b\lambda}(\mathbf{r}) \right] d\lambda \quad (4.2)$$

where κ_λ is the spectral absorption coefficient, $I_\lambda(\mathbf{r}, \Omega)$ is the spectral intensity of radiation (radiance) which is a function of the position $\mathbf{r}$ and direction Ω , and $I_{b\lambda}(\mathbf{r})$ is the black body function given by Planck's law. The radiative transfer equation (RTE) must be solved [6] for the radiation field $I_\lambda(\mathbf{r}, \Omega)$.

There are extensive discussions of the approximate and numerical procedures that have been adopted to solve the radiative transfer equation [6]. The discrete ordinates approximation method (DOM) is adopted to solve the RTE as a compromise between relative simplicity of computer programming, compatibility with the finite-difference method for solving the thermal energy equation, accuracy and computer resources requirements. In the DOM the entire solid angle is discretized in a finite number of directions to which weight factors are associated. The RTE is then solved along each ordinate and the integrals over all solid angles are approximated by a quadrature scheme based on the moment-matching method. Such a method is referred to as the S_N -approximation, where N is the order of the approximation (i.e., the number of values for the direction cosines considered within the range ± 1). Recently, a number of two- and three-dimensional models for simple geometries have been described and they are reviewed by Modest [5]. It has been shown by Fiveland [19] that the S_4 -approximation is a good compromise between accuracy and computation time.

The integration over the spectrum indicated in equation (3) is approximated using the sum-of-gray-gases model [20], and the divergence of the radiative flux vector $\nabla \cdot \mathbf{F}$ is replaced by

$$\nabla \cdot \mathbf{F} = 4\sigma T^4 \sum_n \kappa_n a_{g,n}(T) + \sum_n \kappa_n \int_{\Omega=4\pi} I_n(\mathbf{r}, \Omega) d\Omega \quad (4.3)$$

where $a_{g,n}(T)$ are weighting factors representing an integral over a pseudo-wavelength band where κ_n is constant and $a_{g,n}(T)$ has the property that

$$\sum_n a_{g,n}(T) = 1 \quad (4.4)$$

The weighting factor $a_{g,n}(T)$ represents the fraction of blackbody radiation over some wavelength band n where the absorption coefficient is characterized by κ_n . The values of κ_n and $a_{g,n}(T)$ used in the calculation were taken from Smith et al. [14].

For a gas/soot mixture the absorption coefficient for the gas κ_n is replaced by κ_{n+s} where

$$\kappa_{n+s} = \kappa_n + \kappa_s \quad (4.5)$$

The mean absorption coefficient of soot κ_s is approximated by

$$\kappa_s = C f_v T \quad (4.6)$$

where C is an empirical constant, f_v is the local soot volume fraction, and T is temperature.

As already indicated, the RTE is solved using the discrete ordinates method and $I_n(\vec{r}, \Omega) = I_n^m$, which for a two-dimensional geometry can be written as

$$\mu_m \frac{\partial I_n^m}{\partial x} + \xi_m \frac{\partial I_n^m}{\partial y} = -\kappa_{n+s} \left[I_n^m - I_{bn}(T) \right] \quad (4.7)$$

For discrete directions m , the values μ_m and ξ_m are the direction cosines of Ω_m .

The boundary conditions, say, at the crown, $y=H$, can be written as

$$I_n^m = \epsilon_n I_{bn} + (\rho_n/\pi) \sum_{m'} w_{m'} \left| \xi_{m'} \right| I_n^{m'}, \quad y=H \quad (4.8)$$

where w_m are weight functions in direction $-m$. The first term on the right-hand-side of equation (9) accounts for emission and the second for diffuse reflection of radiation. The boundary conditions at the other boundaries can be written in a similar manner.

Once the radiation field $I_n(\mathbf{r}, \Omega) = I_n^m(\mathbf{r})$ has been determined, the second term on the right-hand-side of equation (4.3) can be calculated

$$\sum_n \kappa_n \int_{\Omega=4\pi} I_n(\mathbf{r}, \Omega) = \sum_n \kappa_n \sum_m w_n I_n^m(\mathbf{r}) \quad (4.9)$$

where w_m is the quadrature weighting factor [5].

Solution of equation (4.7) for each of the m directions are iterative, because the boundary conditions depend on intensity I_n^m . The RTE, equation (4.7), is finite-differenced and is solved by point-by-point iteration. The numerical solution procedure described in the literature [5,9] is followed and the details need not be repeated here.

The solution of RTE and determination of the divergence of the radiative flux vector and of the radiative fluxes at the boundaries were compared against published results [21,22] and good agreement has been obtained. This has provided confidence in the DOM for predicting radiative transfer in an absorbing-emitting medium confined in a rectangular enclosure.

4.3. Results and Discussion

4.3.1 Scope of Parametric Studies

The furnace model considered is the same as that analyzed in Section 3. This is done so that the results already obtained can be used as reference, and the input of radiative transfer in the axial direction on the thermal performance on the model furnace can be assessed.

The large number of parameters describing the furnace include the following:

1. Furnace design and materials (see Fig. 4.1)

- Height H
- Length L
- Radiative properties of refractories
- Crown refractories and their thicknesses

2. Furnace operating conditions

- Specific firing rate, SFR
- Oxidizer preheating temperature
- Fuel burnout length
- Load (glass) surface temperature
- Stoichiometry

3. Soot conditions

- Soot volume fraction at inlet to furnace $f_{v,o}$
- Soot burnout length.

An attempt has been made to identify the most important model parameters, and parametric calculations were then performed to determine the effect of these parameters on the thermal performance of the combustion system. Before presenting the results of parametric calculations a baseline simulation has been performed and the results are discussed to obtain feel for the thermal performance of the system. The results obtained should not be considered as absolute, but should be taken as relative.

4.3.2 Baseline Simulation

The model parameters for the furnace design and operating conditions chosen were somewhat arbitrary but are considered to be representative of a glass melting furnaces. They were as follows:

Height, $H = 1.5$ m

Length, $L = 10.0$ m

Crown emittance, $\epsilon_c = 0.6$

Overall crown heat transfer coefficient, $U_c = 20$ W/m²K

Glass surface temperature, $T_G = 1800$ K

Specific fuel firing rate, $SFR = 2.0$ MW/m²

Stoichiometric combustion

Preheated oxygen at inlet to furnace, 1500 K

Soot volume fraction at inlet to chamber, $f_{v,o} = 0$.

Fuel burnout length L_b is defined as a length by which 99% of fuel has burned out.

A comparison of the gas temperatures predicted by the one-dimensional (1-D) and the quasi-two-dimensional (Quasi-2-D) models is given in Fig. 4.2. The results show that the gas temperature distribution calculated using the Quasi-2-D model is lower than those calculated by the 1-D model. This is due to the fact that radiation heat losses through the openings in the ends of the furnace have been considered in the Quasi-2-D and not in the 1-D model. It should be mentioned that the thermal conditions of the "environment" (i.e., temperature and the effective emittances of the surfaces) are expected to affect the radiation heat losses from the furnace ends.

A comparison of the local heat fluxes along the load surface for the 1-D and Quasi-2-D simulations is illustrated in Fig. 4.3. The heat flux for the latter simulation is lower than for the

former, because the gas (flame) temperature along the model furnace is lower. The average convective heat flux is less than one percent of the radiative heat flux and can be neglected. For $x/L < 0.05$ the local heat flux is negative because heat is extracted from the gas because the assumed glass surface temperature is high ($= 1800$ K). The strong peak in heat flux is undesirable, as this may cause glass reboiling and shorten crown refractory lifetime [2].

With the heat losses only through the crown, and absence of losses through the sides (ends), the heat transfer effectiveness calculated by the 1-D model is 65%, which is much higher than for regenerative, side-fired glass melting furnaces in practice. This effectiveness is for stoichiometric oxygen to fuel ratio and preheated oxygen air temperature of 1500 K at the inlet to the furnace. The global heat transfer parameters for the model furnace are summarized in Table 4.1. The table reveals that thermal performance of the model furnace predicted by the Quasi-2-D model is poorer than that one based on the more idealized 1-D model. Because of the idealizations made in the analysis, the results obtained should be interpreted with caution. The results based on the Quasi-2-D model are more realistic than those obtained using the 1-D model.

Table 4.1 Comparison of the model furnace thermal performances based on the 1-D and Quasi-2-D models:
SFR = 2.0 MW/m^2 and $L_b/L = 0.4$

Model	$T_{g,\max}$ (K)	$T_{c,\max}$ (K)	$\bar{q}$ (MW/m ²)	η (%)
1-D	2625	2114	0.195	65.0
Quasi-2-D	2569	2093	0.152	50.2

Results of sensitivity calculations revealed that the local radiation heat flux at the glass surface was not very sensitive to the emissivity of the crown and end wall refractories. For

example, results of calculations showed that increasing the effective emissivity of the furnace end walls from 0.4 to 1.0 changed the heat flux at the glass surface at most by $\pm 5\%$ and only near the end walls.

4.3.3 Effect of Soot Volume Fraction

The effect of soot volume fraction at the inlet to the furnace $f_{v,0}$ on the gas temperature and load heat flux distributions are shown in Figs. 4.4 and 4.5, respectively. The results reveal the expected trends. As the soot volume fraction is increased the peak gas temperature decreases, and simultaneously the heat flux to the load surface increases as a consequence of the increased opacity of the gas + soot mixture. Note that for $x/L > 0.5$ the gas temperature approaches and then drops slightly below the load (glass) temperature of 1800 K. This is owing to the radiation heat loss through the openings at the ends of the furnace ($x/L = 1.0$). It should be noted that for the simulations performed the soot in the combustion products was not allowed to burnout. Effect of soot burnout will be considered in the next subsection of the report.

The effect of soot volume fraction at the inlet of the furnace $f_{v,0}$ on the thermal performance of the model furnace is summarized in Table 4.2. The results show that the peak gas temperature decreases with $f_{v,0}$, whereas the peak crown temperature increases with $f_{v,0}$. This is consistent with expectations and the trend agrees with the results detailed in Section 3 using the 1-D plug flow thermal energy transport model. Note that the average heat flux $\bar{q}$ and the heat transfer efficiency η appear to "saturate" rapidly with the soot volume fraction $f_{v,0}$ at about $f_{v,0} = 1 \times 10^7$ and remain practically independent of $f_{v,0}$ for larger values. The results of Figs. 4.4 and 4.5 also suggest that the fuel burnout length $L_b/L = 0.4$ is too short, and a longer flame is expected to enhance the thermal performance. The results reported in Table 3 reveal that the presence of soot in the combustion products can enhance the heat transfer rate to the

load in the range of 26 to 29% for the thermal and operating conditions and furnace design considered. The relative magnitude is consistent with the results reported Section 2, but the absolute heat fluxes are lower because of heat radiation loss through the openings at the furnace ends. Results of sensitivity calculations revealed that changing the effective ambient temperature to which the inlet port of the furnace was exposed affected the gas temperature inside the furnace up to the peak, but beyond the peak the gas temperature was not influenced. The port size and the surroundings to which the furnace interior is exposed through the ports would have to be properly modeled to obtain realistic predictions of the furnace performance.

Table 4.2 Effect of soot volume fraction at the inlet on the thermal performance of the model furnace predicted using the Quasi-2-D model: $SFR = 2.0 \text{ MW/m}^2$

$f_{v,o}$ (-)	$T_{g,max}$ (K)	$T_{c,max}$ (K)	$\bar{q}$ (MW/m ²)	η (%)	$\bar{q}/q_o$ (-)
0	2569	2093	0.152	50.2	1.0
1×10^{-7}	2376	2264	0.196	65.1	1.29
2.5×10^{-7}	2356	2319	0.195	65.0	1.28
5.0×10^{-7}	2351	2327	0.192	63.9	1.26
10.0×10^{-7}	2350	2327	0.191	63.7	1.26

4.3.4 Effect of Soot Burnout Length $L_{b,s}/L$

A series of simulations were carried out where the gas burnout length was fixed at $L_b/L = 0.4$ and the soot burnout length was varied parametrically. The gas temperature distribution along the furnace, and the heat flux variation along the load are illustrated in Figs. 4.6 and 4.7, respectively. The results do not reveal a strong effect and show that the local heat release due to combustion controls the gas temperature and heat flux distributions. The variation of soot volume fraction along the furnace does not have much effect either on the gas temperature or the heat flux variations.

The global thermal performance results of the furnace models are summarized in Table 4.3. The table reveals that the performance is not sensitive to the soot burnout length for values of $L_{b,s}/L \geq 0.4$ under the operating conditions calculated. The findings are consistent with the results based on the plug flow model.

Table 4.3 Effect of relative soot burnup length $L_{b,s}/L$ on the thermal performance of the model furnace: $SFR = 2.0 \text{ MW/m}^2$, $L_b/L = 0.4$ and $f_{v,o} = 2.5 \times 10^{-7}$.

$L_{b,s}/L$ (-)	$T_{g,max}$ (K)	$T_{c,max}$ (K)	$\bar{q}$ (W/m ²)	η (%)	$\bar{q}/\bar{q}_o$ (-)
0.2	2488	2144	0.165	51.8	1.09
0.4	2370	2277	0.188	62.7	1.24
0.6	2360	2303	0.193	64.5	1.27
0.8	2358	2310	0.194	64.7	1.28

4.3.5 Effect of Simultaneous Fuel and Soot Burnout Length

The simultaneous burnout of fuel and soot was studied and its effect on gas temperature and heat flux distributions is shown in Figs. 4.8 and 4.9, respectively. The more gradual conversion of chemical to sensible energy in the model furnace reduces the peak gas temperature and makes the distribution much more uniform. Simultaneously, the peak heat flux to the load is also reduced, and the distribution becomes much more uniform. This is desirable from the furnace operation and lifetime point of view.

The global thermal performance results are summarized in Table 4.4. The results show that the average heat flux $\bar{q}$, thermal effectiveness η and the average heat flux ratio $\bar{q}/\bar{q}_o$ are affected little by the relative burnout length as long as all of the fuel and soot are burned. From the heat transfer effectiveness point of view, the model furnace should be made shorter (i.e., less

than $L/H < 10/1.5$) to reduce the heat losses through the roof (crown) as long as all of the fuel is burned. The lower average soot volume fraction in the furnace decreases the average heat flux and the heat transfer effectiveness only little.

Table 4.4 Effect of relative fuel and soot burnout lengths
on the thermal performance of the model furnace:
 $SFR = 2.0 \text{ MW/m}^2$ and $f_{v,o} = 2.5 \times 10^{-7}$

$L_b/L = L_{b,s}/L$ (-)	$T_{g,max}$ (K)	$T_{c,max}$ (K)	$\bar{q}$ (MW/m ²)	η (%)	$\bar{q}/q_o$ (-)
0.2	2684	2573	0.189	63.0	1.24
0.4	2370	2277	0.188	62.7	1.23
0.6	2232	2151	0.188	62.6	1.23
0.8	2152	2079	0.186	61.9	1.22

4.4. Summary

The one-dimensional plug flow analysis described in Section 3 of this report for predicting the thermal performance of a high-temperature glass melting furnace has been extended by developing a quasi-two-dimensional (Quasi-2-D) model in which radiative transfer in both the transverse and longitudinal directions have been accounted for. Based on the results of parametric calculations performed, the following conclusions can be drawn:

1. The largely parametric model (i.e., neglect of turbulence, mixing and chemical kinetics, one-dimensional slug flow, specification of fuel and soot burnup lengths, etc.) cannot be expected to yield realistic predictions of gas temperature and heat flux distributions along the furnace. The model can be used, however, to assess the effects of individual parameters and trends in thermal performance.
2. In spite of the much greater effort needed for analysis, computer programming and computer time required to generate the results, the Quasi-2-D did not reveal any

unexpected results and trends which were not predicted by the simpler plug flow model.

3. The specific firing rate, the relative fuel burnout length and the soot volume fraction are parameters of first order importance which impact the thermal performance of the model furnace analyzed.
4. The soot volume fraction at the inlet to the furnace is an important operating parameter, which for the range of fractions considered, was shown to enhance the heat transfer rate to the load from about 22 to 29% compared to a furnace in which soot is absent.

Figure 4.1 Schematic of plug flow, quasi-two-dimensional furnace model.

Figure 4.2 Comparison of gas temperature distributions along the furnace for the baseline simulations based on the 1-D and Quasi-2-D models.

Figure 4.3 Comparison of the heat flux distributions at the load surface for the baseline simulations based on the 1-D and Quasi-2-D models.

Figure 4.4 Effect of soot volume fraction at the furnace inlet on the gas temperature distribution along the furnace predicted using the Quasi-2-D model: SFR = 2 MW/m².

Figure 4.5 Effect of soot volume fraction at the furnace inlet on the heat flux distribution along the load predicted using the Quasi-2-D model: SFR = 2 MW/m².

Figure 4.6 Effect of relative soot burnup length $L_{b,s}/L$ on the gas temperature distribution along the furnace predicted using the Quasi-2-D model: $SFR = 2.0 \text{ MW/m}^2$, $L_b/L_l = 0.4$ and $f_{v,0} = 2.5 \times 10^{-7}$.

Figure 4.7 Effect of relative soot burnup length $L_{b,s}/L$ on the heat flux distribution to the load predicted using the Quasi-2-D furnace model: $SFR = 2.0 \text{ MW/m}^2$, $L_b/L = 0.4$ and $f_{v,o} = 2.5 \times 10^{-7}$.

Figure 4.8 Effect of relative fuel and soot burnup length on the gas temperature distribution along the furnace predicted using the Quasi-2-D model: $SFR = 2.0 \text{ MW/m}^2$ and $f_{v,o} = 2.5 \times 10^{-7}$.

Figure 4.9 Effect of relative fuel and soot burnup length on the heat flux distribution along the furnace predicted by the Quasi-2-D model: $\text{SFR} = 2.0 \text{ MW/m}^2$ and $f_{v,0} = 2.5 \times 10^{-7}$.

5. CONCLUSIONS AND RECOMMENDATIONS

Parametric analyses have been developed to predict radiation heat transfer from a luminous flame to glass in a model glass melting furnace using natural gas as fuel and industrial oxygen as the oxidizer. A systematic approach has been followed by first starting from a very simple "well-stirred" furnace model and developing successively more realistic models by relaxing some of the idealizations introduced originally. Based on the results obtained the following conclusions can be drawn:

1. The flame length (i.e., coverage of glass surface by flame) is an important furnace design and operating parameter. The longer the flame the better is the furnace thermal performance. For example, the effective flame length can be increased by the burner design or staged combustion of fuel.
2. The soot volume fraction at the inlet to the combustion chamber impacts significantly the furnace thermal performance. For a soot volume fraction of 1×10^{-6} the heat transfer effectiveness can be increased by about 25 compared to that in the absence of soot in the fuel.
3. The presence of soot lowers the peak flame and crown temperatures, but the maximum heat flux is not reduced in spite of lower flame temperature owing to the increase in the flame emittance caused by the presence of soot.
4. A large number of idealizing assumptions have been made to simplify the analysis. The predictions need to be validated by comparing the results with test data or at least against results of more detailed computational models.
5. A first principles based combustion/heat transfer model is needed to realistically predict the thermal performance of the furnace. Based on the recent computational advances,

the thermal performance of the furnace. Based on the recent computational advances, such modeling appears to be feasible, but significant effort would be required to develop a more detailed model (i.e., including models for soot oxidation, chemical kinetics, radiative transfer, turbulent chemically reacting and radiating flow, etc.) and write a computer program for solving numerically the model equations.

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