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MASTER

ENERGY MANAGEMENT IN RESIDENTIAL AND SMALL COMMERCIAL BUILDINGS

ANNUAL REPORT, FISCAL YEAR 1976

July 1976

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ANNUAL REPORT, FISCAL YEAR 1976

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July 1976

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Abstract

The goal of the present program is to develop the technical basis for efficient energy use in space heating of residential and small commercial buildings. Part One of the report describes efficiency measurements performed on conventional residential oil-fired hot water heating equipment, including both steady state and cyclic (part load) efficiency determinations. A list of preliminary recommendations for retrofit actions to improve efficiency is provided at the end of Part One. A summary of work carried out in the areas of thermal storage media, fenestration, and building thermal dynamics is presented in Part II.

INTRODUCTION

Space heating is a major energy-consuming function in both residential and commercial buildings. In 1970, 20% of the energy consumed in the United States was used in residences and 15% in commercial buildings. More than half of the residential demand, or 11% of the total energy used in the U.S., was allocated to space heating and in the commercial sector space heating accounted for 7% of the national total. Space heating is the largest component of energy use in buildings and is the most likely candidate for savings in the residential and commercial sectors.

The objectives of the present program are to provide the technical basis for reduced consumption of oil and gas and to promote the efficient use of all energy sources in space conditioning of residential and small commercial buildings. Work covered includes improvements in specific devices, methods of interfacing devices with building structures, and formulation of control philosophies for optimum management of thermal energy flows in buildings. Thermal energy storage at ambient temperatures in the building structure (and in auxiliary thermal storage devices, where appropriate) is a central design feature that provides for optimal utilization of hybrid energy sources, including solar energy and off-peak electrical energy from coal and nuclear sources.

The current program has been divided into the following tasks:

Task 1 - Program Management Plan

Task 2 - Performance Efficiency Measurements for Space Heating Devices

Task 3 - Development of Instrumentation Techniques for Laboratory and
Field Measurements

Task 4 - Engineering Design, Testing and Analysis of New Heating Systems
and Controls

Task 5 - Thermal Energy Storage in Building and Auxiliary Devices

Task 6 - Thermal Storage Media

Task 7 - Fenestration and Heat Transfer Control

Task 8 - Building Thermal Dynamics, Simulation and Analysis of Integrated
Space Conditioning Systems

Task 9 - Burner-Boiler and Furnace Testing, Evaluation, and Analysis

Part One summarizes the progress made under Tasks 2, 3, and 4, described
below.

Task 2: Determine the efficiency of residential heating appliances in
both field and laboratory studies. Provide data for the determination of the on-
off cycling characteristics of home heating appliances. Evaluate the effect of
on-off cycling on the overall efficiency of heating units. Develop a base line
of efficiency data for evaluating various modifications in terms of improvement
in overall efficiency.

Task 3: Develop specialized measurement techniques to monitor the unsteady
(cyclic) efficiency of residential heating systems. Consider such techniques as
the enthalpy flow method. Investigate other applicable techniques that become
available in the course of laboratory investigation. Develop one or more of
these techniques for use in the field in determining the overall efficiency of
home heating units, including the effect of cycling. Develop a device for use
by the homeowner and service personnel that will offer quantitative feedback on
the operating efficiency of the space heating system. Investigate and demon-
strate the cost effectiveness of such a device.

Task 4: Develop, test, and analyze heating systems and controls for both new and retrofit heating systems of improved operating efficiency. Develop and test new heating units incorporating many energy conservation features, such as a low-mass, regeneratively cooled, thermally purgeable furnace. Develop improved control system efficiency. Permit heating system performance to conform more closely to the needs of residents. Improvements to be considered shall include features such as variable aquastat temperature limits, reduced nozzle size to lengthen burner cycles, and variable temperature-set points during the day.

Part Two summarizes the work performed under Tasks 6, 7, and 8.

PART ONE: OIL-FIRED HOT WATER HEATING EQUIPMENT

I. DISCUSSION

Traditionally, the performance of residential and commercial heating units has been evaluated at full-load operation. However, in field installations heating units operate at part load and steady-state evaluation is no longer valid. Recently it has been established that the actual fuel utilization efficiency in the field is well below the level determined by the steady-state value. A major goal of this program is to quantify the loss in efficiency resulting from cyclic (part-load) operation of conventional heating units and to identify procedures for improving their overall thermal efficiency.

The current designs of fossil-fuel-fired heating units owe more to historical practice than to modern technological capabilities. Such devices generally consist of fuel injector and burner which fires into a properly sized combustion chamber. Adjacent to the combustion chamber, heat transfer by induced and forced convective heat exchange between the combustion gases and the thermal medium is achieved in, e.g., a boiler heat exchanger. A common version of such a unit is the oil-fired hot-water boiler provided with a "tankless" domestic water heater

to allow space and domestic water heating from a single unit. We may define several "efficiencies" for such a device.

Combustion efficiency: The combustion efficiency characterizes the effectiveness of the combustion process in converting the chemical enthalpy of the reactants (fuel and air) to the sensible enthalpy of the combustion products. This efficiency is virtually unity for a properly operating burner of modern design.¹ Combustion efficiency is defined by

$$\eta_{\text{comb}} = \frac{\text{Combustion product enthalpy rise}}{\text{Equilibrium product enthalpy rise}}.$$

Accordingly, for a properly maintained installation, we may take $\eta_{\text{comb}} = 1$.

Steady-State Efficiency - The steady-state efficiency is a measure of the effectiveness of the heating unit in extracting the enthalpy for the hot combustion products and transferring it to the medium (e.g., water) used for space heating when the entire system is operating in a steady-state mode. This implies that the unit has been operating, after ignition, long enough to establish steady-state temperature profiles everywhere in the system. The steady-state efficiency may be defined as

$$\eta_s = \frac{\text{Steady state enthalpy removal rate thru the boiler}}{\text{Steady state chemical enthalpy injection rate}}.$$

Depending on the characteristics of the particular space heater, η_s values may vary widely. Steady-state efficiencies generally fall within the range $0.45 < \eta_s \leq 0.85$.

Overall efficiency: In a typical boiler installation the combustion device operates only a small fraction, typically 20%, of the time. The boiler is

usually a massive device subject to long cooling periods during which its stored enthalpy undergoes substantial losses. It follows that over many cycles (e.g., during the heating season) the steady-state efficiency is substantially greater than the overall efficiency, which represents the fraction of useful heat available for space heating. Accordingly, we may define the overall efficiency, $\bar{\eta}$, as

$$\bar{\eta} \equiv \frac{\int \text{Enthalpy removal rate thru boiler } dt}{\int \text{Chemical enthalpy injection rate } dt}$$

where the integrals are taken over very long times to obtain a good statistical number of cycles. In a typical case, $\bar{\eta}$ is about 50 to 60%.

Cycle efficiency: It follows from the above that the relative magnitude of these "cycle losses" can be described in terms of a cycle efficiency, η_c , defined as $\eta_c = \bar{\eta}/\bar{\eta}_s$. Typical current installations vary widely, but in all cases the burner is oversized relative to the peak space heat load requirements. Consequently, (1) the burner is "on" for only a small fraction of the time, typically 20%; (2) the typical "on" time is short, reflecting the ability of the burner to quickly heat the boiler to temperature, i.e., the firing rate is overdesigned; and (3) at maximum load conditions, the burner usually does not approach continuous operation.

II. SUMMARY AND CONCLUSIONS

The enthalpy flow measurement technique was used to measure the thermal efficiency of oil-fired hot-water heating units under steady-state and cyclic operation. In steady-state operation, conventional stack-gas sampling measurements were found to overestimate the fraction of heat transferred from the fuel oil combustion products to the boiler water. During cyclic operation, the

efficiency of the boiler decreased as a function of the percent of time the burner was "on" (heat load). Examination showed that as the fuel-firing rate was increased, the steady-state efficiency decreased.

Field performance data indicate that the "average" burner is "on" ~ 20 to 25% of the time. In the test boiler, such "on" times resulted in off-cycle thermal losses of 15 to 20%. The overall efficiency of the test unit during cyclic operation was 60 percent compared with a steady-state efficiency, determined by stack-gas sampling of 83%.

Preliminary recommendations for retrofit actions to improve efficiency are given in Section V.

III. EXPERIMENTAL

A. Laboratory Investigation

1. Objectives

One important area of the program is the measurement of the fuel utilization efficiency of oil-burning residential heating equipment. A laboratory has been established at BNL to make such measurements on various residential boilers. The work now includes the determination of both steady-state (continuous operation) and cyclic performances. The measurements are based on absolute enthalpy flow, which can be used as a standard method for evaluating a wide range of heating units.

The BNL laboratory investigation is directed toward quantitative determination of the degradation of the overall efficiency of heating units. A relationship has been demonstrated between the cycling times of current burners and the efficiency of heating system operation.^{2,3} The BNL examination attempts to generalize this relationship for many different types and arrangements of heating units and for many different burner cycles. In addition to the effect

of varying the "on" time of the burner, the effect of varying the cycle length and firing rate is under investigation.

The operating efficiency is determined by monitoring the change in enthalpy of the working fluid (water or air) as the fluid passes through the heating unit. The overall operating efficiency is then defined as the total fraction of chemical enthalpy (fuel heating value) converted to sensible heat in the working fluid.

Specialized measurement techniques are used to monitor the cyclic (unsteady) efficiency of residential heating units. In this investigation, new techniques, suitable for use in the field, will be considered. A device with this function would provide the homeowner and service personnel with quantitative feedback on the operating efficiency of the space heating system.

An important part of the laboratory investigation is to provide base-line data on the operation of heating units. Proposed retrofit modifications will be tested in the laboratory against this standard and a quantitative evaluation made of improvements in efficiency.

2. Laboratory Facilities at BNL

a. Heat Rejection Equipment. Removal of the heat generated during efficiency tests can be accomplished in a once-through flow system, where the heated water is removed from the system, or in closed-loop operation, where heat removal is facilitated by a water-to-air heat exchanger.

In the once-through mode of operation an elevated storage tank is used for the cold inlet water to remove pressure fluctuations from the supply line and to provide a constant inlet pressure to the boiler. The water is heated as it passes through the boiler. It then flows to a drain that removes it from the system. This method of operation eliminates heat feedback between the outlet

and inlet of the boiler. The inlet water can be held at a constant temperature, regardless of the outlet water temperature. This simplifies the analysis of system performance, particularly during cyclic operation.

In closed-loop operation, the water from the boiler outlet flows to a water-to-air heat exchanger, where a variable-speed fan controls the rate of heat rejection from the water. The cooled water is returned to the inlet of the boiler and reheated. The main advantage of the closed-loop system is that it more closely simulates boiler operation in a home. The water flow rates, circulator running times, and heat load of the heat exchanger (fan speed) can be varied to simulate the heat load found in residences. The temperature of inlet water to the boiler is no longer constant but can vary in response to the system load.

b. Exhaust Systems. To provide for a variety of stack conditions, three exhaust systems for flue gases were constructed.

(1) Brick chimney. A brick chimney of the type and dimensions commonly found in homes was built at the Laboratory. The draft-inducing effect is a function of height and flue-gas temperature and will depend on the heating and cooling rates and the thermal capacity of a particular design. In performance, this chimney closely simulates those found in field installations. This factor is most important during the cyclic (part-load) evaluation of heating systems.

(2) Sheet metal flue. To eliminate the thermal capacitance effect of chimneys, a sheet metal flue was constructed. Some modified heating system designs to be tested may not require a negative stack pressure.

(3) Induced-draft system. An induced-draft fan with appropriate ducts and dampers was constructed and installed to provide a continuously vari-

able negative stack pressure from zero to one-half inch water column. This system can be used to establish any draft condition desired during the testing program.

c. Instrumentation.

(1) Water flowmeter. The enthalpy flow measurement technique requires determination of the mass flow rate of water through the boiler. A turbine-type water flowmeter is used for this purpose. It is a high-frequency device with magnetic pickup and the appropriate electronics to convert frequencies of volumetric flow rates.

(2) ΔT - Change in water temperature through the boiler. A thermopile provides for accurate determination of the ΔT across the boiler. This device, consisting of two 2-in. pipes, each containing an array of 10 thermocouples, measures the temperature differential of the water flowing to the inlet and from the outlet of the boiler.

(3) Oil flowmeter. The oil flow is monitored by a low-flow turbine-type flowmeter similar to that used to determine water flow. Oil-flow measurements are also carried out by volumetric (buret) and gravimetric (analytic balance) techniques. The mass flow rate of the oil in pounds per hour is determined.

(4) Fuel-oil heating value. The heating value of the fuel oil (Btu per pound) is determined by using an oxygen bomb calorimeter in conjunction with the appropriate ASTM standards. These values are compared with handbook values for the fuels being utilized. With calorimeter, a wide range of fuels for use in home-heating systems (e.g., methanol) can be examined.

(5) Thermocouples. Copper-copper constantan thermocouples are placed at various locations in the test system to follow the temperature-time history

of the boilers. Ice baths are used when accurate measurements of temperature levels are required.

d. Data Acquisition and Analysis. At present the outputs from the temperature and flow transducers are recorded on chart paper and paper tapes. Data analysis requires manual integration of areas under curves and the use of planimeters and other time-consuming techniques.

In steady-state operation the data analysis is simplified, since the system is time invariant and the temperatures and flow rates approach constant values. During cyclic operation the analysis becomes more difficult and tedious. Integrated values for the oil mass flow and the product of temperature and water mass flow are required. The only method now available involves manual techniques.

To make operation more efficient and precise, a complete digital recording-integrating system is being built at BNL. This system will be capable of automatic data acquisition and reduction, including the integration of appropriate mass flow and enthalpy flow values. Variable time-scale printers and displays will be available to give the appropriate time rate and integrated flow and enthalpy values. One channel will indicate the accumulated efficiency value.

e. Test Units. The laboratory investigation is focusing on an assortment of commonly available off-the-shelf oil-fired boilers. The selection of particular equipment has been arbitrary, but the equipment under examination represents the type of heating equipment commonly found in homes. The equipment currently being tested in the laboratory includes

(1) Vertical steel fire-tube hot-water boiler with low mass combustion chamber and high-speed motor designed for a firing rate of 1 gph.

(2) A cast-iron sectional hot-water boiler with low mass combustion chamber and high-speed motor designed for a firing rate of ~ 1.25 gph.

(3) A horizontal steel fire-tube hot-water boiler designed for outdoor installation with fuel firing rate of up to 1 gph.

(4) A down-fired steel hot-water boiler design for reduced cycle losses with a new flame retention head burner firing at about 2 gph.

3. Test Procedure

The enthalpy flow measurement technique is used to evaluate both the steady-state efficiency and the cyclic efficiency of the heating system being tested.

a. Steady-State Efficiency. The steady-state efficiency indicates the fraction of chemical enthalpy from the combustion process that is converted to sensible enthalpy (temperature rise) in the heat transfer fluid (water). Determination of the steady-state efficiency by the enthalpy flow technique requires measurement of the mass flow rate of the water, the rise in temperature of the water as it passes through the boiler, the heat capacity of the water, the mass flow rate of fuel oil to the burner, and the heating value of the fuel oil. From these values a ratio can be established representing the fraction of heat transferred to the water for steady-state operation (see Figure 1). The system is operated for at least one hour before data are recorded, to allow all the flow rates and temperatures to reach steady time-invariant levels. At the same time that enthalpy flow measurements are recorded, the steady-state efficiency is measured by conventional flue-gas sampling methods. The corrected stack-gas temperature and the percentage of CO_2 in the flue gases are recorded and the corresponding steady-state efficiency is determined. The efficiency obtained

EFFICIENCY CALCULATION

HEAT INTO WATER = $M_w C_w \Delta T$ (C_w = HEAT CAPACITY)

TOTAL OIL HEAT = $M_{oil} H_{oil}$ (H_{oil} = HEATING VALUE)

$$\text{EFFICIENCY } (\eta) = \frac{M_w C_w \Delta T}{M_{oil} H_{oil}}$$

Figure 1. BNL laboratory equipment schematic.

by this method can be compared with the actual efficiency determined by the enthalpy flow method.

b. Cycle Efficiency. For the once-through flow method of heat rejection, cyclic performance is evaluated by two modes of operation:

(1) Pulsed cycle. The system is brought to a reference temperature. The combustion process is then initiated and continues for a predetermined period. After burner operation has ended, the system remains idle for a predetermined period. Finally, the heat available from fuel combustion is determined by the enthalpy flow technique.

(2) Continuous cycle. In this case, a digital electronic cycle programmer is used. The preselected "on" and "off" times of the burner are maintained at constant values. The system is operated until the transient effect (due to the initial temperature of the heating system) is damped out. Heat is removed by once-through water flow. Once a "steady cycle" is obtained, the fuel input and heat output for one or more cycles are evaluated. These data are used to determine the overall thermal efficiency of the heating system.

In both cases, the steady-state operating efficiency is measured immediately following the cycle test to take into account daily barometric fluctuations, which have a small but measurable effect on the efficiency of the systems.

Once the corresponding steady-state and overall efficiencies are determined, a value can be obtained for the cyclic performance of the system:

$$\eta_{\text{cycle}} = \frac{\eta_{\text{overall}}}{\eta_{\text{steady state}}}$$

B. Field Investigation

1. Objectives

A field data collection program has been established by BNL to determine the cyclic performance of residential heating systems and provide data for the analysis of the on-off cycling characteristics of home heating units. Guided by these findings the BNL laboratory investigation is evaluating the effect of on-off cycling on the overall efficiency of heating units in the field. From these base-line data, various modifications to improve overall heating efficiency can be evaluated. The field study includes the installation of passive monitoring equipment in selected homes to determine the total operating time and cycling rate of the burner and circulator. Analysis of these data yields values for the total percentage of time that the burner operates. The average time of each on-off cycle can be determined on a daily, monthly, or yearly basis.

2. Equipment and Procedure

The on-off cycling of home heating units is monitored with cycle counters and timers. The timers provide the total elapsed running time for the burner and circulator at each installation. The counters indicate the total number of on-off cycles. This information can be used to calculate burner "on" time percentages and the average length of each cycle. Chart recorders are used to obtain more detailed cyclic information (e.g., variations in the length of each cycle).

Ten Long Island homes have been equipped with these devices. Residents take readings once or twice a day and enter them on standard data sheets. The performance of the burners and circulators is then evaluated. The resulting data include

Average "on" time per cycle $\frac{\text{Burner "on" time}}{\text{Number of cycles}}$

Fractional running time of burner $\frac{\text{Burner "on" time}}{\text{Total elapsed time}}$

Average length of each cycle $\frac{\text{Total elapsed time}}{\text{Number of cycles}}$

Hourly temperature data are used to calculate the average outside temperature between readings for each data point. A plot for each hour of burner "on" time versus the average outside temperature provides useful information regarding the seasonal performance of the heating systems under varying load conditions.

IV. EXPERIMENTAL RESULTS AND DISCUSSION

A. Laboratory Investigation

1. Steady-State Efficiency Measurements

The enthalpy flow measurement technique was used to determine the steady-state efficiency of the cast-iron and the vertical steel fire-tube boilers. The resulting values were used as a basis on which to compare corresponding steady-state efficiencies obtained by conventional stack-sampling methods (using commercially available equipment). Table 1 summarizes the results of tests conducted over a period of several months.

Table 1

Steady-State Efficiencies (%)

	Enthalpy flow measurement	Stack-gas sampling method	Difference (%)
Cast-iron boiler	71.8	78	8.6
Vertical steel fire- tube boiler	77.1	83	7.6

In all cases, the stack-gas sampling procedure gave steady-state efficiency values ~8% greater than those determined by enthalpy flow measurement. This discrepancy is beyond the limits of experimental uncertainty. Since a large fraction of the field efficiency data collected by various investigators was obtained with use of stack-gas sampling techniques, a more detailed and broader investigation should be conducted to reconcile this difference in measurement techniques. Currently available field performance data should be re-evaluated to provide more accurate efficiency values.

Although the data in Table 1 indicate that the fire-tube configuration is more efficient than the cast-iron boiler, this is not a valid interpretation. The efficiency is a function of many variables, including the type of burning equipment, the fuel firing rate, and the percentage of excess combustion air (stoichiometry of the flame). In the two systems tested, different burners were used for the two heating units. In addition, the fire-tube boiler was equipped with a 1.0 -gph fuel nozzle, while the cast-iron boiler used a 1.2 -gph nozzle. Tests indicate that the steady-state efficiency increases as the fuel firing rate is reduced (see Figure 4). Also, the burner mounted on the steel fire-tube boiler was operated with less excess air than the cast-iron unit, which would improve the relative performance of the fire-tube boiler.

The enthalpy flow measurement technique provided consistent, reproducible results. The heating systems efficiencies varied in response to daily barometric conditions (affecting the mass flow rate of flue gases). These fluctuations could be monitored and consistent average values were obtained for the steady-state performance of the heating units being tested. The efficiency values determined for the once-through flow system and the closed-loop mode of operation were in close agreement.

2. Cycle Efficiency Measurements

The cyclic performance of the steel fire-tube boiler was investigated by the enthalpy flow technique, with use of the once-through flow system and pulsed operation.

To simulate the performance of field heating units, the test boiler was "detuned" to a steady-state efficiency of 72%, an average value for field units.⁴ Efficiency degradation was accomplished by increasing the amount of excess air entering the system at the combustion chamber. After each cycle test, the steady-state efficiency was measured to monitor fluctuations during the cycle tests.

Figure 2 is a plot of cycle efficiency versus burner "on time" percentage. Qualitatively, this curve is similar to results obtained by others.^{2,3} As the heating load is reduced and the burner "off" time is increased, a larger fraction of available heat is subject to off-cycle flue losses. Consequently, the overall heating system efficiency is reduced by intermittent operation.

At a burner "on" time of 20%, the cycle efficiency is about 80%. With a steady-state efficiency of 72%, the overall heating system efficiency becomes 58%. In other words, a large fraction of the heat available from the fuel oil is lost as a result of burner "on" and "off" cycle losses.

The efficiency of operation drops rapidly as the burner "on" time decreases below 20%. The use of hot-water boilers to supply domestic hot water results in small burner "on" times during off-heating-season months. Field data indicate that burner operation to satisfy the domestic hot water load alone averages 5%. At this "on" time, the cyclic efficiency falls below 40%, for an overall heating unit efficiency of ~25%. For summer operation three-fourths of the heat available from the oil is lost in supplying domestic hot water.

Figure 2. Cycle efficiency versus $\frac{\text{burner "on" time}}{\text{burner "off" time}}$
percentage $\eta_{ss} = 0.72$ burner "on" time = $14 \frac{1}{3}$ min.

The cycle performance data presented as the total heat loss vs burner "off" time in Figure 3 lead to a physical explanation of the "hyperbolic" behavior of the curve in Figure 2. For the cyclic mode of operation, the total heat loss from the boiler is related linearly to the burner "off" time (τ_{off}). As time increases, heat is lost and the temperature of the storage device (the boiler) decreases. This is indicated by a gradual decrease in the slope of the curve with increasing "off" time, showing a reduction in the heat loss rate for the lower boiler temperatures. For "off" times less than several hours, the heat loss rate is linearly related to τ_{off} . Therefore, a plot relating heat loss (or efficiency) and $1/\tau_{\text{off}}$ should be hyperbolic in shape. This is indeed the case in Figure 2.

The hyperbolic shape can be explained in more physical terms. From Figure 3, we can assume that the heat loss rate is constant. As the burner "on" time (τ_{on}) is reduced from 30 to 20 to 10%, τ_{off} does not increase linearly. Assume a τ_{on} of 10 min. For τ_{on} of 30%, τ_{off} is 23 1/3 min., for 20%, it is 40 min, and for 10%, 90 min. Since we assume the heat loss rate to be linear with τ_{off} , the longer burner "off" times correspond to larger heat losses. As a result, the curve presented in Figure 2 is highly nonlinear as smaller loads are approached because the length of the "off" cycle increases nonlinearly.

Future tests will examine the effects of boiler temperature, the length of cycles, and other parameters on the cyclic efficiency of heating units. Steady cycle operation will be used instead of pulsed operation to more closely simulate the performance of heating units in the field.

Since at present only new units are being tested in the laboratory, the older equipment in the field is expected to exhibit lower steady-state and

Figure 3. Total heat loss (10³ Btu) versus burner "off" time (min.).

cyclic efficiencies. The burners being tested are of the high-speed flame retention type. Therefore, the percent CO_2 in the flue gases (an indication of steady-state efficiency) is expected to be higher for these test units than for the average field unit. Field tests by the State University of New York at Stony Brook, the Environmental Protection Agency and others indicate an average percentage of CO_2 of $\sim 8\%$ by volume. The average efficiency for these field tests is $\sim 72\%$ for steady-state (full-load) operation.

The older field units are expected to exhibit a poorer cyclic performance also. The cycle losses incurred during the burner "off" time are related to the thermal mass of the heating systems. The older heating systems are generally more massive than the modern units, with massive combustion chambers, commonly constructed of firebrick. The large amount of heat that can be stored in these heavy combustion chambers contributes directly to "off" cycle losses and reduced cycle efficiency. In the more modern heating units the combustion chamber consists of preformed, low-mass materials with only a small fraction of the heat capacity of firebrick.

The cycle performance of heating systems is affected by the "off" cycle flue flow, which in turn is affected by the type of burner equipment (forced draft or natural convection), flow resistance of the boilers, and the magnitude and location of air leaks into the heating unit. All these factors must be considered before laboratory results can be applied to field installations.

3. The Effect of Reduced Firing Rates on Steady-State Efficiency

Preliminary laboratory tests to determine the effect of reduced firing rate on the steady-state efficiency of hot-water boilers were conducted on the steel fire-tube boiler in the once-through water flow mode. Fuel nozzles of

varying sizes were installed and the burner was adjusted to give a bacharach smoke number of 1.

As can be seen in Figure 4, a reduction in the firing rate results in an increase in the steady-state efficiency of the particular system tested. The oil flow rate was reduced from 1.28 to 0.678 gph, with a resulting increase in steady-state efficiency from 74.1% to 78.2% -- an increase of 5 1/2%.

Note that, while the efficiency increased, the percentage of CO₂ in the flue gas decreased as the firing rate was reduced. The criterion used for burner adjustment, after each nozzle change, was a smoke number of 1. The decrease in percent CO₂ as the burner firing rate was reduced indicated that more excess combustion air was required to achieve the same smoke number. This degradation of burner performance at lower fuel flow rates was the result of less turbulent mixing of the fuel and air when the burner inlet air damper was closed. Reducing the combustion air pressure drop available at the burner head led to poorer burner performance. If the CO₂ had been maintained at a constant value (instead of the smoke number) the increase in steady-state efficiency would have been larger than 5 1/2%.

The increase in steady-state efficiency can be attributed to a reduction in the volume flow rate of the combustion products through the heat exchanger (boiler). The heat exchange volume is constant, and at lower flow rates the residence time of the hot gases on the heat exchange surfaces is increased. As a result a larger fraction of useful heat can be transferred to the working fluid to increase the efficiency of operation. The reduction in combustion-product flow rate through the boiler can be achieved by reducing the excess combustion air or reducing the fuel firing rate. In either case, consideration

Figure 4. Steady-state efficiency versus oil flow rate (steel fire-tube boiler).

must be given to the effect of these changes on other performance criteria such as the rate of soot formation.

4. Variation in Fuel Nozzle Flow Rate

During the laboratory tests it was observed that the fuel flow rate through the nozzle decreased monotonically with time and approached a constant value during steady-state operation. Figure 5 shows the flow rate versus time for a 0.75 gph nozzle from a cold start. The fuel rate decreased by ~9% over a 30 min. period. The nozzle flow rate for intermittent operation was also observed (see Figure 6). The variation from the steady-state flow value was a function of the burner "off" time. Two curves are presented in Figure 6 for 5-min and 10-min burner "off" times following steady-state operation.

Physically, the fuel nozzle flow rate appears to be a function of temperature, decreasing as the temperature increases. During steady-state operation the fuel injection hardware is subjected to several heat fluxes. Heat is supplied by radiative energy transfer from the flame and combustion chamber walls. Convective cooling takes place through an external flow of cool combustion air and an internal flow of fuel oil. These processes compete to establish a steady operating temperature and steady flow rate. The long time-scale required to establish equilibrium (~30 min) indicates that thermally massive system components are involved in this equilibrating process.

During intermittent operation, the convective cooling flows cease as burner operation is terminated. The energy stored in the combustion chamber radiatively heats the fuel injection equipment above steady operating temperature. As the burner is restarted, the temperature of the nozzle and associated hardware is above its equilibrium value and the oil flow rate is below its steady value. The competition of the heating and cooling modes, together with the length of the

Figure 5. Dimensionless oil flow rate (from cold start) versus burner "on" time, 0.75 gph nozzle.

Figure 6. Oil flow rate versus burner "on" time (for intermittent operation), 1.0 gph nozzle.

"on" and "off" cycles, determines the subsequent fuel flow performance. For long "off" periods, more time is required to return to the equilibrium flow value.

This variation in fuel flow rate during intermittent operation raises questions regarding the performance of heating systems in the field. Field units exhibit "on" times of ~10 min. In a strict sense, the system never reaches a steady-state oil flow value. The magnitude of the variation in fuel flow will depend on the length of burner "on" and "off" times. While the oil flow variation may be only 2 or 3% during typical operation, it is clear that the concept of steady-state evaluation of heating system performance requires more careful examination.

B. Field Investigation

The field data collection program was initiated in February 1975. Cycle performance data were collected and analyzed for the latter part of the 1974-1975 heating season. A plot of the burner percent "on" versus average outside temperature was prepared for each home. From the average values for all the homes monitored (see Figure 7) it can be seen that at an outside temperature of 40°F, the burner operates <20% of the time. At 30°F the burner operates between 25 and 30% of the time. A weighted seasonal average for the burner "on" time would be expected to fall in the 20% to 25% range.

The curves in Figure 8 display the relationship between burner "on" time and average outside temperature for five homes. The solid data points represent the homes with known nozzle sizes. At an outside temperature of 30°F the home with the 0.75 gph nozzles had burner "on" time 3 times as great as those for the home with the 1.25 gph nozzles. The homes were in no way matched with respect to heat loads or system efficiencies and this merely reflects a

Figure 7. Burner percent "on" (average values) versus average outside temperature (°F).

Figure 8. Burner "on" time (%) versus average outside temperature.

tendency for the homes with smaller nozzles to exhibit longer burner "on" times, as expected.

Actually, the home with the 0.75-gph nozzle is atypical. If data from this home are eliminated and an average burner "on" time is estimated from the data presented in Figure 8, at 30°F the burner "on" time falls between 20 and 25%.

A reasonable minimum design temperature for this locality is 10 to 15°F. At this temperature most burner "on" times fall below 50%, an indication that the heating units tested are 100% oversized, which would contribute to the short "on" times during most of the heating season. Oversizing of heating equipment has been observed by many investigators.

Data collected from ten homes for one calendar year are currently being analyzed at BNL for use in determining average seasonal performance values for local residential heating equipment.

The cyclic efficiency of heating units determined by laboratory investigation and the cycling characteristics of units in the field can be used to evaluate overall heating unit efficiency. The efficiency of the system at a specified percentage of "on" time and the relationship between "on" time and outside temperature can be combined to determine an overall heating unit efficiency for a particular locality and specified heating equipment.

V. PRELIMINARY RECOMMENDATIONS FOR RETROFIT ACTIONS

TO IMPROVE EFFICIENCY

BNL has compiled a list of retrofit actions for improving the efficiency of oil-fired heating units (see Table 2). The energy savings per retrofit represent average values estimated from the BNL laboratory study and current liter-

Table 2*

Retrofit action	Estimated cost (\$/unit)	Total potential market (10 ⁶ units)	Estimated max. annual retrofits (10 ⁶ units/yr)	Energy savings per retrofit (10 ⁶ Btu/yr)	Payback period for oil @ 40¢/gal (yr)
1. Reduced burner nozzle size	20	11	11.00	18.7	0.37
2. Retrofit controls					
(a) Variable aquastat for boiler water temperature	150	14	3.90	18.7	2.78
(b) Timers for thermostat control	100	11	3.66	16.6	2.10
(c) Zone controls	150	5	1.00	10.4	5.00
(d) Continuous circulator operation	30	10	5.00	10.4	1.00
3. Burner steady-state efficiency adjustment	20	14	14	5.2	1.33
4. Retention-head burner with reduced firing rate	150	11	2.75	25	2.10
5. Blue flame burner	200	13	3.25	25	2.78
6. Variable firing-rate burner	300	12	2.00	31.2	3.33
7. Convective stack damper to reduce downtime losses	150	11	2.75	20.8	2.50
8. Outside air ducting for combustion air	100	7	2.33	12.7	2.77
9. Purgeable, low mass boiler	400	7	0.37	41.6	3.33
10. Water-fuel emulsion	150	13	2.16	12.7	4.16

Notes: (1) Savings from retrofit actions are not additive. (2) Options 6 to 10 have less development.

*Based on a total of 14×10^6 single family units and oil consumption of 500×10^6 bbl/yr.

ature. The values presented and the impact of each retrofit action will change as various retrofit modifications are developed and examined.

A. Reduced Burner Nozzle Size

One of the most effective and easily implemented retrofit actions involves the reduction of burner fuel nozzle size. Both steady-state and cycle efficiencies are improved as a result of reducing the firing rate. The improvement in steady-state efficiency is discussed in Section IV.A.3. With a smaller nozzle size the cycle efficiency will also increase (see Section IV.A.2), since the burner will operate for a larger fraction of the time. Recent field data demonstrate that average reductions in firing rates of ~25% reduce fuel consumption by ~7%.^{5,6}

There are practical limitations on the amount that the fuel firing rate can be reduced in home heating units. Burner performance can be adversely affected by a reduction in firing rate, since less effective mixing of the atomized fuel and combustion air increases the likelihood that smoke will form. In addition, the use of "tankless heating coils" to supply domestic hot water restricts the amount that the fuel nozzle can be reduced. In this case the rate of heat production determined by the rate of domestic hot-water use dictates a minimum nozzle size of 1.1 to 1.2 gph. While the space heat load can be satisfied with a much smaller nozzle, the rate of domestic hot water use requires a large fuel flow rate.

A simple solution to this problem promises significant reductions in fuel consumption for combustion devices supplying both space heat and domestic hot water. The installation of a domestic hot-water storage tank will eliminate the need for large, instantaneous fuel firing rates. The tank can be heated gradually through a tankless coil at the reduced firing rate, and the stored hot water can

satisfy peak demand. The tank must be well insulated and thermally isolated from the boiler during the burner "off" cycle.

During the heating season the efficiency of operation is improved by using a reduced fuel firing rate to supply both space heat and domestic hot water. Fuel savings can also be made available during off-season operation to provide domestic hot water. As already noted, the use of a boiler with a tankless coil to supply domestic hot water during the summer leads to overall heating system efficiencies of <25% (see Section IV.A.2). This is a direct result of maintaining the boiler at a constant high temperature to satisfy the infrequent demand for hot water. An insulated storage tank eliminates most of this thermal loss. Instead of cycling the burner all day to maintain the boiler temperature, the burner is operated for one long cycle per day. During this "on" cycle the storage tank is "charged" to supply the daily hot water demand. The boiler is allowed to return to ambient temperature and no further losses are attributed to boiler "off" cycle losses. The efficiency of operation approaches the steady-state value.

B. Retrofit Controls

Several modifications in heating system controls that could lead to substantial fuel savings are being considered, and "consumer-oriented" control package incorporating many energy-saving features is being developed at BNL. Some of these control modifications are listed in Table 2.

1. Variable Aquastat Temperature for Hot Water Systems

Varying the aquastat setting would control the temperature of the boiler water. Off-cycle losses are proportional to the boiler water temperature, and reduced water temperatures yield reduced off-cycle losses. For peak load conditions, a "maximum" boiler temperature is required. For smaller heating loads

the aquastat temperature (boiler water temperature) could be reduced. This should increase the overall efficiency of the system.

2. Timers for Thermostat Control

Timers for controlling the thermostat set point can be used for nighttime temperature setback, to reduce total heating requirements. It has been estimated (Honeywell, Inc.) that energy savings of about 8% can be achieved in this way. In addition, the timers can be used to reduce the thermostat setting during working hours, vacations, and other times when the structure is vacant. This allows the heating system controls to be programmed to meet the comfort requirements of the inhabitants.

3. Zone Controls

The temperatures of separate living areas in a structure can be controlled individually by use of zoning. The total heat load can be reduced by lowering temperatures in uninhabited areas. Careful analysis of the thermal dynamics of the structure is required to ensure proper operation of the zone controls for maximum energy savings. The impact of zoning is estimated in Table 2.

4. Continuous Circulator Operation

Continuous operation of the circulator provides minimum boiler temperatures during the burner "off" cycle. A common control arrangement for oil-fired hot water systems links the thermostat to the water circulator at the boiler. As the temperature in the home falls, the thermostat contacts close and the circulator begins transporting boiler water to room radiators. When the boiler water temperature drops below a low limit setting, the aquastat contacts close and the burner is energized. Thus, the heat load controls the thermostat, the thermostat controls the circulator, and the circulator (through the aquastat) controls the burner. In this mode of operation the boiler water is maintained between a high

and a low limit setting and average boiler temperatures of 170° to 200°F are common.

In another mode of operation, continuous circulator operation provides for lower average boiler temperatures and reduced off-cycle thermal losses. In this case the thermostat controls the burner directly. The circulator continuously purges heat from the boiler so that the average boiler temperature is held to a minimum value. In actual installations the circulator could be de-energized at an appropriate low boiler temperature limit of 90° to 100°F. The actual low limit set point must be determined by optimizing the heat delivery and the electrical energy consumption of the circulator.

Conservative values for energy savings resulting from continuous circulator operation are given in Table 2. These values will be adjusted as additional testing data become available.

C. Burner Steady-State Efficiency Adjustment

The role of burner equipment is to mix the fuel and combustion air in the proper proportions for efficient combustion. The steady-state efficiency of the burner-heat transfer equipment is a function of the percent CO_2 in the flue gases and the net stack temperature (see Figure 9). The steady-state efficiency increases with an increase in the percent of CO_2 in the flue gas. As the stack temperature increases, a larger fraction of heat is lost up the flue in the form of sensible enthalpy of the combustion products, and the steady-state efficiency decreases.

The most efficient operation corresponds to the maximum percent CO_2 and the minimum stack temperature. A typical average value of steady-state efficiency is indicated by the dashed line in Figure 9.

Figure 9. Efficiency versus percent CO₂ at various corrected stack temperatures (°F).

The theoretical percentage of CO_2 available by stoichiometric (zero excess air) combustion of fuel oil is ~15%. This value is rarely attained in combustion equipment. In real devices non-homogeneous mixing of the fuel droplets with the combustion air pattern requires excess combustion air for completion of the combustion process. The size distribution of the atomized oil droplets enhances this nonhomogeneous performance.

The excess air needed for complete combustion of the fuel serves to decrease the efficiency of operation. Figure 10 relates the percent excess air to the percent CO_2 in the flue gases. As the excess air percentage is increased, the volume percent of CO_2 in the flue gas is reduced and the steady-state efficiency decreases. This process can be explained by a simple energy balance. The combustion air enters the burner at ambient temperature (70°F). After the combustion-heat transfer process the gases exit at an elevated temperature (500 to 600°F). The result is a net heat loss.

The stack temperature is also affected by excess combustion air. As the mass flow rate of combustion products increases through the boiler, the stack temperature rises and the efficiency of operation is reduced. For the most efficient adjustment of a heating unit, excess combustion air used must be a minimum. The main limitation on this minimum is the formation of smoke and incomplete combustion products. As the excess air flow is reduced, smoke (particulates), carbon monoxide, and unburned hydrocarbons tend to increase exponentially. These present obvious environmental problems. Also, the soot accumulates on the heat transfer surfaces and reduces the heat transfer coefficient and the steady-state efficiency. Therefore, the combustion process must be adjusted for the maximum percent CO_2 consistent with an acceptable level of smoke formation.

Figure 10. Percent CO₂ versus excess air
for combustion of No. 2 fuel oil.

Table 2 indicates that adjustment of the burner steady-state efficiency effects a modest reduction in fuel consumption. It has been estimated^{3,5} that on a national basis an average increase in efficiency of 2% can be achieved by adjustment of the steady-state efficiency.

D. Retention-Head Burner With Reduced Firing Rate

The installation of a burner with retention-head capable of producing at least 12 percent CO₂ with a smoke number of 1 would substantially increase the steady-state efficiency of conventional heating systems. The reduction of excess air requirements leads to improved burner efficiency (Section V.C.). The selection of a burner designed for operation at low firing rates (<0.75 gph) ensures optimum performance. The improvement in efficiency for reduced firing rates is discussed in Section V.A.

The combination of retention-heat burner and reduced firing rate provides the capability for an estimated minimum improvement in efficiency of ~15%.

E. Blue Flame Burner

Blue flame combustion makes it possible to maintain both a high percent CO₂ in the flue gases and an acceptable smoke number. As a retrofit modification the blue flame burner requires additional development. The blue flame devices now on the market require a precisely sized combustion chamber matched to a specific fuel firing rate. This arrangement severely limits the use of blue flame burners in retrofit installations. The blue flame system is expected to give improvements in efficiency of the same order of magnitude as the reduced firing-rate retention-head burner modification. Quantitative evaluation of blue flame burners is required to substantiate their capability for fuel improvements.

F. Variable Firing-Rate Oil Burner

Variable firing-rate burners are under consideration as a means of improving the overall efficiency of oil-fired heating equipment. Improved fuel injection techniques would allow firing rates below .5 gph.

The variable firing-rate burner provides longer burner "on" times for the entire heating season. For moderate heating loads, the burner could operate at a low firing rate. As the heating load increases the burner firing rate could be increased to satisfy peak demands. The effect of a variable fuel-firing rate is to increase the total burner "on" time and reduce the off-cycle losses. Much developmental work is required to make this retrofit option a viable alternative.

G. Convective Stack Damper To Reduce Downtime Losses

Installing a stack damper is an obvious method for reducing off-cycle flue losses. At the end of each burner "on" cycle the damper closes to eliminate off-cycle flue flow. The safety of such a device is a consideration. Once the flue is closed, any fumes or combustion products in the system are trapped. In a conventional system, the chimney effect tends to purge the system during the "off" cycle.

Fuel nozzle "after-drip" is common in conventional oil burners. After flame-out, fuel is delivered to the combustion chamber as the burner motor winds down to a halt. This unburned fuel is vaporized as it enters the hot combustion chamber. Once the stack damper is closed, these fumes are trapped in the heating unit. Carbon monoxide produced by the burner shutdown transient could be trapped in the heating unit and constitute health hazard for inhabitants of the home. Also, the accumulation of combustibles could lead to an explosive gaseous mixture in the heating unit.

The installation of a damper in the combustion-product flue flow exposes the damper materials to gases of relatively high temperature during burner operation. The failure of such a stack device to open during any burner operation could create a dangerous situation. Therefore, the mechanical damper device must perform precisely at each "on" cycle.

At BNL, the concept of a flow damper located at the inlet to the burner is being developed. The damper is subjected to ambient-temperature air flow -- an advantage over the stack flow damper. The inlet damper is passive in design and opens in response to the negative pressure produced by the combustion air fan. This particular design allows for a controlled purge flow through the heating unit during the off-cycle. In any case, consideration must be given to safety hazards resulting from the installation of any flow damping device.

H. Outside Air Ducting for Combustion Air

The flow of combustion air through the heating unit contributes to the infiltration heating load of the structure. Ambient room air (at 70°F) consumed by the combustion device is replaced by outside air (at a lower temperature). This additional heat load (estimated as 5% of the total heat load) must be supplied by the heating unit. The use of outside air for combustion eliminates this portion of the heat load. Using outside air should result in only a small reduction in steady-state efficiency. From this point of view, outside air ducting can significantly reduce the total heat load of the structure. At the same time, it can increase off-cycle flue losses. The off-cycle loss rate is a function of the temperature differential between the boiler and the ambient air flowing through it during the off-cycle. More heat will be lost to cold ambient air flowing through the boiler during a particular "off" cycle than to air at 70°F.

A more detailed examination is required to quantify the net effect of outside air ducting on overall fuel utilization efficiency.

I. Thermally Purgeable Low-Mass Boiler

The concept of a high efficiency, purgeable, low mass boiler is under development at BNL (see Figure 11). This device consists of two separate sections, a combustion heat transfer section and a hot-water storage reservoir. The purpose of separating the combustion and thermal storage functions is to minimize "off"-cycle thermal losses and achieve cycle efficiencies approaching unity.

The combustion-heat transfer section is of low-mass construction for reduced heat storage. It is designed for thermal purging (removal of stored heat) at the start of the burner "off" cycle. This keeps the combustion-heat transfer section at or near ambient temperature during the burner "off" cycle, and "off"-cycle flue losses are negligible. The useful heat is stored in the thermally insulated hot-water tank.

J. Water-Fuel Emulsion

The use of water-fuel emulsions can improve oil burner operation by reducing the amount of excess air required for "complete" combustion. In order to burn, fuel oil must be atomized -- separated into a cloud of small droplets. The droplets are mixed with the combustion air and burn in a semi-premixed manner. As a result of incomplete combustion of the larger droplets, which require longer retention times through the "burning zone" to react completely, smoke and combustion intermediates are produced. To facilitate more complete combustion, additional air (above the stoichiometric value) must be supplied by the burner. This excess air reduces the steady-state efficiency of the heating unit (see Figures 9 and 10).

Figure 11. High efficiency purgeable low mass boiler.

Emulsion burning improves the efficiency by reducing the amount of excess air required for complete combustion. As the emulsion droplets enter the combustion chamber they are heated radiatively by the hot surroundings. The water inside the oil droplet vaporizes and the droplet breaks apart. This process (secondary atomization) enhances mixing of the fuel oil and combustion air and reduces the mean oil-droplet diameter. The result is a reduction in excess combustion air and a corresponding increase in combustion efficiency.

The penalty incurred is the loss of energy required to heat and vaporize the water. This is reflected in lower adiabatic flame temperatures for oil-water emulsions. The net energy savings can be determined by subtracting the energy needed to prepare and burn the emulsions from the improvement in efficiency resulting from reduced excess combustion air.

Surveys of the performance of residential heating equipment indicate excess combustion air levels of ~80% (steady-state efficiencies of ~72%). In this case, the excess air levels are very high and the use of fuel oil-water emulsions may reduce fuel consumption. A detailed energy balance is needed to determine the advisability of using a fuel oil-water emulsion and to establish the optimum mass ratio of water to fuel oil.

PART TWO: ENERGY MANAGEMENT IN BUILDINGS

A summary of the work carried out under Tasks 6, 7, and 8 is given below.

I. THERMAL STORAGE MEDIA (TASK 6)

The following is an extract of a preliminary report on a search for possible thermal storage media.⁷

A. Introduction

The first-order phase transition occurs with a discontinuity in any of the first derivatives of the free energy G , such as the entropy S , the volume V , and

the heat content H . In the case of melting, discontinuities exist in ΔS , ΔV , and ΔH at the transition temperature. The change in the heat content, ΔH (namely, the heat of fusion) is a quantity that can be reversible, depending on the direction of change in temperature. Consequently, a system can be devised to give up, or adsorb, this amount of heat, ΔH , and this becomes an energy storage system.

Such a system requires a material medium whose solid-liquid transition temperatures range between 65° and 75°F (18 and 23°C). The material should be low in cost, available in large quantities, and easy to handle.

B. Material Types

Materials of many types were considered. Binary metallic systems were rejected for two reasons, one being the high cost of metal alloys. To meet melting point specification (18° to 23°C), alloys must contain expensive elements such as In, Ga, and Na. The second reason is the difficulty of handling the liquid metallic alloys, which are notorious for causing liquid embrittlement of metals.

Binary halide systems are also eliminated on the basis of high cost. In addition, they are usually explosive in solid form. They are used as solid propellants in many cases. Binary oxide systems often do not have eutectic temperatures within the temperature specifications. Those that do meet the specification usually contain an oxide of metal ions in an excited state (or unusual valent state), and the compounds are rather unstable.

Initially, the most attractive candidates seemed to be the numerous liquid crystal materials. The phase transitions from solid to liquid crystal states vary with the generic type. This variable can be adjusted according to the system specifications. As the search progresses, it is apparent that liquid crystal falls short on three counts. First, the transition temperature range is usually

higher than the specified range. Second, the supply of liquid crystal materials is always very small. Third, the materials are very expensive.

Attention was next turned to natural compounds that exhibit liquid crystal behavior, including some of the natural soap materials such as oleate, palmitate, and laurate. These materials have melting points that are higher than required. Fortunately, the melting points decrease in mixtures. In addition, the mixtures are commercially produced as commercial-grade fatty acids and their derivatives.

C. Fatty Acids

At this stage of the investigation, fatty acids and their derivatives seem to be attractive material candidates because their melting points are within the temperature range, the heat of fusion is reasonable in amount, they are relatively inexpensive (about 28¢/lb), and they are available in large quantities and can be supplied on a carload basis.

Some of the more attractive materials are listed in Tables 3 to 7. There are, however, several questions requiring further study.

One concerns the flammability of these materials. From the available data, the flash points are usually $>200^{\circ}\text{C}$.

Second is the miscibility of these materials. It would be advantageous if the stearamide (Kemamide S, Table 7) could be mixed with lower m.p. materials such as oleic acid (red oil, Industrene 105, Table 7). The extremely low cost of Kemamide S (without pelletizing, it costs only 5.65¢/lb) is most attractive. But the question is whether the melting point of the mixture will be sufficiently below that of Kemamide S (m.p. $\sim 136^{\circ}\text{C}$).

Third is the question of odor, since many of these materials are produced from either animal fats or vegetable oils.

Table 3

Material	Trade name	Titer (°C)	Carload price (¢/lb)
Fatty acid (95% capric)	Hydrofol Acid 1095	28 -31	65
Soya fatty acid (water white dis- tilled)	Veg Acid 120	21 -24	46.75
Linseed fatty acid (regular distilled)	Veg Acid 505	17 -24	47.25
Linseed fatty acid (water white dis- tilled)	Veg Acid 520	17.5-20.5	57.5
Isobutyl stearate	Starfol BS120	17 -21	45

*Supplier: Chemical Products Division, Ashland Chemical Company, (Division of Ashland Oil Inc.).

Table 4

Material*	Trade name	Titer (°C)	Carload price (¢/lb)
Polyethoxylated fatty acid	Ethofat 242		70.75
Polyethylene glycol ester	PEG 600 mono- oleate	23	64
Polyethylene glycol ester	PEG 600 diolate	19	60
Polyethylene glycol ester	PEG 600 mono- laurate	23	68
Polyethylene glycol ester	PEG 600 dilaurate	18	72

*Supplier: Industrial Chemicals Division, ARMAK.

Table 5

Material*	Trade name	Max Titer (°C)	Carload Price (¢/lb)	Present use
High-purity commercial oleic acid	Pamolyn 100	12	43	chemicals
Food-grade oleic acid (Kosher approved)	Pamolyn 100FG	13.5	46	food
Oleic acid	Pamolyn 125	12	34.5	soap
Conjugated linoleic acid	Pamolyn 347	5	42	resin
Tall oil fatty acid	Pamak 4	15	27	soap
Fatty acids	Pamak W4	10	27	soap
Fatty acids	Pamak WD	10	25.5	soap
Fatty acid	Pamak 4A	10	28	industrial

*Supplier: Organic Department, Hercules Inc.

Table 6

Material*	Trade name	Titer (°C)	Carload price (\$/lb)
Soya fatty acid, alkyd grade	Emery 614	18-23	31
Soya fatty acid, super alkyd grade	Emery 618	18-23	33
Tallow fatty acid	Emery 880	10-20	29
Hydrogenated methyl tallowate	Emery 2204	30	33
Methyl myristate	Emery 2214	17	27.5
Methyl oleate	Emery 2219	18	28

*Supplier: Emery Industries.

Table 7

Material*	Trade name	Titer (°C)	Carload Price (\$/lb)
Oleic acid (red oil)	Industrene 105	205	32
Distilled cotton fatty acid	Industrene 138	32-38	28
Distilled animal fatty acid	Industrene 143	39-43	18
Stearamide	Kemamide S	136	5.65

*Supplier: Chemical Division, HUMKO Products.

Finally, there is the question of thermal hysteresis in the phase transition. The possible degrees of supercooling and superheating must be investigated in these materials.

No statistics are available on current production yields. However, since some of these materials are derived from animal fats for vegetable oils and are used in food, paints, etc., their availability should not be a problem.

Furthermore, for this specific application, lower purity grades of these materials can be used, with a concomitant reduction in material costs.

Other materials will be evaluated in subsequent phases of this investigation.

II. FENESTRATION (TASK 7)

At light-transmitting elements of structures (architectural windows, skylights, solar collector windows, greenhouse fenestration, etc.) three energy transport modes are generally involved: convection of heat, molecular conduction, and radiative transport. To optimize the energy conservation features of such light-transmitting elements (windows), suppression of convective heat transfer within the window elements (e.g., between two panes of glass) is desirable. Unwanted radiative intrusion and/or unwanted radiative losses must be controlled. Where light and/or thermal energy transport through a window are desired, optimum transport by one mode (radiation) generally is to be coupled with optimum suppression of other modes (conduction and convection). Accordingly, for a large number of "window" systems, it is useful to examine methods of separating these energy transport modes.

One such set of approaches is illustrated in Figures 11 and 12 of ref. 8. Another set is illustrated in Figure 12. Here, convection-defeating elements are inserted between two panes of glass. Thin (aluminized) reflector arrays

INTERIOR

EXTERIOR

HINGED SHADE IS LIFTED BY STRINGS,
WHICH CLEAR NOTCHED EDGES OF THE
FIXED SHADE ELEMENTS

Figure 12. Convection inhibiting fenestration
with radiation control.

suppress free convection between the panes, even though high heat and light transmission are largely preserved. Of course, optical fidelity is degraded whenever the array is "in place." Remotely operated (rotated, raised, lowered) radiation-convection control arrays can be employed to meet temporal variations in the characteristic use of a given window.

SUNY-BNL joint efforts on fenestration include

1. Design, fabrication, and testing of various window subsystems and systems for controlled separation of the operative energy transport modes, to provide optimum heat and light control.
2. Development of the theoretical formulations necessary to the rational design and evaluation of these window systems.
3. Emphasis on the use of currently available components (e.g., on-the-shelf electrically operated venetian blind systems) and materials (honeycomb, aluminized Mylar, standard glass, glass coatings, etc.).

III. BUILDING THERMAL DYNAMICS (TASK 8)

Simulation and analysis of the thermal dynamics of building components (including the overall system itself -- building and components) is necessary to identify, examine, and demonstrate the effectiveness of components and strategies of worth and promise.

Work to date includes time-dependent heat transfer calculations for ordinary building components (e.g., masonry walls) as well as for super-heat-capacity materials⁹ as employed in a number of energy conservation strategies. Figure 13 illustrates the superiority of a particular super-heat-capacity structure over one made of ordinary masonry. The time-dependent response of an initially isothermal (70°F), 8-in. walled structure is calculated for two cases.

Figure 13. Temperature-time response of ordinary concrete and concrete containing PCM.

The first involves a structure whose masonry-based material contains some 9% of phase-change materials. The phase-transition temperatures are uniformly distributed in the range 65° to 75°F: that is, 10% of the (encapsulated) phase-change material sustains a phase transition at 65° to 66°F, 10% of the phase transition material sustains a phase change at 66° to 67°F, and so on. The assumed physical properties of the composite material are those of common masonry, but containing fatty acid-like encapsulated phase-change materials.⁷

The second case corresponds to a pure masonry structure.

For each of the two systems considered, a 90°F exterior temperature is imposed on the initially isothermal (70°F) temperature. After 12 hr, ordinary masonry attains an interior wall temperature >82°F (>12°F rise) whereas the super-heat-capacity material has increased to 73°F (a rise of only 3°F). Thermal dynamic simulations and studies for a substantial range of ordinary and composite building materials and structural components are contemplated and in progress.

Additional, existing computer programs (e.g., NBS) designed for extensive building thermal dynamic simulation are to be revised to accommodate features such as "phase-change material composites" and nonconstant specific heat capacities.

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