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A FEASIBILITY STUDY OF THE POTENTIAL FOR THE IMPROVEMENT OF BUNKER HEATERS

Final Report
Contract No. DAHC04-71-C-0038

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Space heaters have a primary function to convert varied fuels into heat with maximum efficiency and as a secondary function of use in cooking food in the U. S. Army. Present heaters do not incorporate the latest combustion technology for safety and improved efficiency. As a result of this program, several recommendations have been presented and possible solutions to present problems are discussed and summarized in this report.

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Space heaters have a primary function to convert varied fuels into heat with maximum efficiency and as a secondary function of use in cooking food in the U. S. Army. Present heaters do not incorporate the latest combustion technology for safety and improved efficiency. As a result of this program, several recommendations have been presented and possible solutions to present problems are discussed and summarized in this report.

FOREWORD

This report was prepared for the U. S. Army Land Warfare Laboratory (USALWL), Aberdeen Proving Ground, Maryland by Dr. Robert H. Essenhigh, Professor of Fuel Science of the Combustion Laboratory, The Pennsylvania State University. This report presents the results of a program conducted under Contract No. DAHC04-71-C-0038 with the U. S. Army Research Office, Durham, North Carolina.

Mr. Frederick M. Drake of the Environment and Survival Branch was the USALWL Project Officer.

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GENERAL INTRODUCTION

Space heaters for bunkers, tents, and huts have a primary function to convert fuel into heat with maximum efficiency, and a possible secondary function for use in cooking. However, the key to any improved design must be based on improving overall efficiency. Sources of inefficiency are as follows:

(1). Poor combustion due to bad design. This allows escape of unburned fuel or fuel derived components, such as carbon monoxide, smoke, etc.: Smoke emission is also undesirable from the military point of view. Total combustion of fuel under natural draft conditions is particularly difficult. Requirements for improving the combustion process are discussed in Part III.

(2). Poor heat transfer between the flame or hot gases to the space to be heated. This is primarily a matter of heat recovery, partly discussed in Part II, to reduce stack losses, but it includes effective utilization of the heat, i.e., the heat demand and viability temperatures discussed in Part I.

(3). Poor utilization of the heat with an incorrect balance between convective and radiative heating. This is discussed in Part I.

(4). Poor combustion control allowing air into the combustor in excess of design requirements. This is jointly a matter of design and operation. Undue excess air increases the stack loss merely by virtue of being "ballast" material that is heated, and discharged with its heat, to no useful effect. It also reduces the stack temperatures and thereby reduces the efficiency of the useful heat transfer processes. This is discussed in Part II.

(5). Undue space loss: excessive loss from the heated space due to:

(i) drawing heated air from inside the bunker, tent, or hut for the combustion process. This air has to be replaced by cold air allowed to inleak (causing drafts).

(ii) poor insulation allowing excessive wall loss.

(iii) poor door design allowing excessive quantities of heated air to escape and cold air to enter as personnel enter or leave the facility.

(6). Poor maintenance that upsets the combustion process due to choked or warped grate, burner, etc.; or allows air inleakage or gas escape as the result of mechanical damage.

Overall, the efficiency depends on reducing all losses to the absolute minimum by adequate design, and keeping them at that level by adequate operation and maintenance.

Safety is an additional design requirement. This has two aspects:

(1). Fire Safety, in respect to which there should be no undue hazard such as naked flames, bare hot spots, or likelihood of fire initiation or spread if the unit is accidentally knocked over.

(2). Health Safety, in respect to which there should be no undue hazard from noxious or poisonous gases (e.g., carbon monoxide) able to escape from the heater in dangerous quantities into the space being heated.

PART I: TARGET HEATING REQUIREMENTS

I.1 Thermal Input - Based on past records, the heater must develop a gross thermal output of approximately 20 Btu per hour per cubic foot of heated space. Thus, in a small tent of roughly 8 ft. cube or about 500 cu. ft. the heat requirement is about 10,000 Btu/hr. For bunkers or huts, the requirements would increase to the range 50,000 to 100,000 Btu/hr. By comparison the estimated heat output from personnel in sedentary or light manual work is of the order of 500 to 1,000 Btu/hr. This is very variable depending on the individual metabolism and the nature of the work, and it rises during eating.

The heat requirements will also vary with: (1) the temperature difference between inside and outside; (2) the surface to volume ratio of the facility being heated (generally decreasing as the ratio increases); and (3) the outside wind velocity (which affects the convective losses). The 20 Btu per hour per cubic foot should therefore be regarded as being only a target figure, with the expectation that it should increase either as the outside temperature drops or as the wind velocity increases.

The target figure of 20 Btu/cu.ft.hr. influences design to the extent that it determines the fuel supply rate and air supply rate (see Part II).

I.2 Expected Efficiency - If efficiency is defined for the purposes here as the fraction of heat that is not lost up the stack (by warm gases and unburned loss), then expected efficiencies could range from 20% to 65%, based on measurements on open fires and freestanding closed stoves using natural draft, reported in the literature. For a properly designed and operated closed-stove, an efficiency in the range 50% ± 10% may be not unreasonable. This, however, will depend in part on the nature of the fuel since a poor burning fuel may require more excess air to ensure complete combustion, and the higher excess air will increase the stack loss.

Consequently, of the 20 Btu/cu.ft.hr. dissipated, about half can be expected to be lost up the stack. In principle, much of this can be recovered, but only at the expense of losing draft with the danger that the fire may go out. This is, however, one initial point where careful examination of draft requirements will be worthwhile.

I.3 Heat Demand - The heat dissipated by the stove supplies the heat demand and this is ultimately lost to the outside by: (1) conduction through the walls of the enclosure being heated; (2) direct loss of heated air through walls, windows, etc., and as combustion air, with replacement by cold air. The heat demand is reduced to the extent that these loss sources can be reduced. The problem of conduction is primarily one of materials and their thickness. This is assumed to be predetermined although a large loss could justify restructuring the standard enclosures using materials with better insulating properties.

A problem in evaluating relative losses is the lack of data. However, when heating houses by stoves with not particularly low outside temperatures (25 to 45°F) a large part of the overall loss is due to combustion air being drawn from the room. To replace this, inleakage must be allowed, otherwise the fire will smoke, and on occasion this will allow further heat loss by exleakage. All evidence is in favor of a separately controlled supply of combustion air to avoid this heat loss. A separate supply would also permit better sealing of the enclosure from the outside without risk of the fire smoking. Provision for tapping a fraction of the air, to maintain freshness, would also be desirable.

Occupants of the enclosure will affect the heat demand primarily by setting the viable temperature. This then sets the temperature difference between inside and outside, and hence the rate of heat loss both by wall conduction and heated air loss. Clearly, the lower the viable temperature the better. The occupants will otherwise have little direct effect on the heat demand once the viable temperature has been set and is being maintained. Their main influence will then be more indirect as the result of moving in and out of the enclosure, thus increasing the losses through the door, and by bringing in wet garments to dry. Since the heat of evaporation of water is nearly 1,200 Btu/lb., which would require about 2,400 Btu additional heat dissipated by the stove at 50% efficiency, wet garments could represent an appreciable addition to the heat demand.

I.4 Viable Temperatures - The temperatures that can be regarded as being at viable levels for the occupants of tents and huts are likely to vary considerably with circumstances. For example, they can be reasonably expected to fall with increasing distance from a base area, in general, and will probably fall with decreasing size of enclosure. Nevertheless, as a generality it can be assumed that viable temperatures will be maintained by the heat dissipation rate of 20 Btu/cu.ft.hr. from the heater at an efficiency possibly in the region of 50% (so the effective dissipation will be closer to 10 Btu/cu.ft.hr.). Any adverse variation in actual air temperature inside the enclosure that cannot be effectively countered by increasing the heat dissipation rate will then be countered, it is assumed, by increased clothing.

However, within the restrictions of a given rate of heat dissipation, the viability of the environment can be improved considerably by a good balance between radiation and convection in heating the enclosure. This was a prime conclusion reached by Fishenden in research conducted nearly 45 years ago (summarized with other data by Himus in "The Elements of Fuel Technology" pp. 452-454). The human body dissipates heat mainly by radiation and by evaporation through the skin. The radiation loss is strongly influenced by the background radiation from walls, windows, stoves, etc.: Skin evaporation is more affected by a combination of humidity and air temperature. Most cold weather heating systems are unbalanced to the extent that air temperatures are generally too high for comfort and radiation levels too low. The air temperature required for comfort can be reduced as the radiation levels are increased. Fishenden found that the most comfortable combination was an air temperature of 55°F with radiation levels sufficient to preserve warmth, at humidity of 40 to 60%. Nonuniform densities of radiation were also found preferable to uniform densities, i.e., the radiating sources should be hotter than the body on one side of the enclosure and cooler than the body on the other side. Fortunately, this matches the usual pattern of heating by a stove. The results also indicated that temperatures exceeding 60°F tended to produce "slight mental lassitude". At temperatures below 50°F, the subjects reported that the radiation levels then required produced the sensation of "scorched on one side, chilly on the other", a well known sensation to those familiar with the open fire method of heating a house in cold weather.

In an arctic tent the viability temperatures may not match the suburbanites levels of comfort as described by Fishenden. Nevertheless, the conclusions are clear. The heater should be designed to radiate rather than convect (a suitable rate of convection is likely to occur anyway). This emphasis should increase the viability of an enclosure at a given level of heat dissipation by the heater. Conversely, a given level of viability will be possible at a lower rate of fuel consumption, i.e., the heater will be more efficient in terms of overall performance (i.e., use efficiency). The thermal efficiency as defined in Sec. I.2 above should also be improved if increased radiation density allows a lower air temperature since the lower temperature difference between inside and outside will immediately reduce all heat losses. The higher radiation levels should also be more efficient in warming an occupant who has just entered from the outside - heated air alone is notoriously inefficient for such a purpose.

I.5 Summary - (1) The target dissipation figure for the heater is therefore about 20 Btu/cu.ft.hr. (gross) at about 50% thermal efficiency, or about 10 Btu/cu.ft.hr. (net).

(2). The heater should be designed to radiate rather than convect.

(3). To reduce enclosure losses, a separate entry for the combustion air supply should be provided.

(4). To reduce the body radiation losses, it might be advantageous to have reflective coatings on all cold walls of the living enclosure.

PART II: GENERAL REQUIREMENTS FOR HYDROCARBON COMBUSTION

II.1 System - A bunker heater, which is the system of concern, is a metal and/or refractory enclosure in which the fuel (solid, liquid, or gas) is burned in air. The products of combustion are predominantly carbon dioxide, water vapor, nitrogen, and excess oxygen, if combustion efficiency (per cent of total fuel burned) is high. If combustion efficiency is low, there will also be significant quantities of carbon monoxide and hydrogen, and possibly smoke (liquid and solid particulates), methane, higher hydrocarbons, polynuclear aronatics, etc:. The system includes: (1) a stack to the outside to remove the combustion products, and (2) air ports to allow entry of the combustion air. The combustion efficiency is governed by the combustion mechanisms, as discussed in Part III. The thermal efficiency, on the other hand, is governed by the combustion air requirements and the associated stack loss. The air requirements and stack loss are therefore the prime aspects of the system behavior to be examined in this section (Part II) of the Study.

II.2 General Approximations and Assumptions - Although the fuels that could possibly be considered for use in the bunker heater are solid, liquid, or gaseous, some useful generalizations exist that enable all hydrocarbon fuels to be treated simultaneously so far as air requirements and stack loss are concerned. To a first approximation we may assume:

(1) That the stoichiometric air requirement for hydrocarbon fuels is about one cu.ft. (at s.t.p.) per 100 Btu released (actual or potential);

(2) That the mass of air supplied for combustion is approximately the mass flowing out again as combustion products and excess air;

(3) That the specific heats of air and combustion products are approximately the same, and do not vary significantly with temperature, for the temperature ranges of concern (a convenient but not essential assumption).

These approximations allow calculation of thermal efficiency and so forth to within ± 10 or 20%. This accuracy is more than adequate for the purpose in hand.

II.3 Actual Air Requirements: Total and Specific - (1) Total Air Required - From the approximation No. 1 given above, the total air supply rate will range from 100 cu.ft. per hour for the small tent to 1000 cu.ft. per hour for a larger bunker or hut at a stoichiometric supply rate. At 50% excess air, which is generally more realistic for natural draft units, the volumes become: 150 to 1500 cfh respectively. The relevant volume of air then becomes combustion products, expanded by anything from 2 to 5 times the original volume (depending on the exit temperature of the gases), and is expelled via the stove pipe or stack. The potential influence of the stack design and dimensions will be self-evident. The point is elaborated in a later section.

(2) Specific Air Requirements - The specific requirement is the air volume to be supplied per unit volume of heated space. At 20 Btu per hour per cubic foot of space, the air requirement is 0.2 cu.ft. per cubic foot for a stoichiometric supply, or 0.3 cu.ft. per cubic foot at 50% excess.

(3) Rate of Air Changes - The specific air requirement is particularly pertinent when related to the frequency of air changes. If the combustion air is drawn from inside the enclosure being heated, then every cubic foot of air in the enclosure will be removed and replaced by (colder) air from the outside once every three hours or so. The thermal loss that this represents due to having to reheat the air volume from the outside ambient temperature to the inside viable temperature is estimated below. The conclusion is that it is always desirable to provide ducting for the combustion air to the stove or heater direct from the outside, to cut down the replacement losses as far as possible.

II.4 Possible Thermal Efficiencies - Thermal efficiency is defined in Sec. I.2 as the percentage fraction of heat that is not lost up the stack; i.e. it is the gross quantity of heat released within a combustion chamber minus the stack loss and unburned loss. The stack loss is that due to dry combustion products loss and moisture loss. The thermal efficiency as defined here is also known as the "Available Heat". It can be calculated quite easily on the basis of the approximations listed in Sec. II.2, but conveniently it has been calculated and presented as a graph in the North American Combustion Handbook, p. 56 (N.Am. Mfg. Co., Cleveland, Ohio: First Ed. 1952, 1957). The graph is reproduced as Fig. II.1 in this report. It applies approximately to all hydrocarbon fuels.

To use the graph, it is necessary to postulate (e.g. for design), or to measure, the flue gas temperature and excess air. Given these, the available heat or thermal efficiency can be read off. At this point it is convenient to emphasize the opposing requirements for excess air and flue gas temperature:

(1) For maximum thermal efficiency, the excess air should be as low as possible, preferably zero, but the demands of combustion efficiency (Part III) always sets a finite value. In practice, 50% excess is usually regarded as typical for a natural-draft, closed stove. (With an open fire it can be as high as 400 to 500%). From Fig. II.1, 50% efficiency is achieved with 50% excess air if the flue gas temperature is 1400°F.

(2) For maximum thermal efficiency, the exiting flue gas temperature should be as low as possible. However, as it drops, the draft - and therefore the drawing power of the fire to pull in the necessary air and use it effectively - also drops. With falling draft, more excess air may be necessary for complete combustion so there is a partial trade off here. Unfortunately, the precise requirements are difficult to estimate through lack of data. Nevertheless, it will be clear that maintenance of some minimum flue gas temperature (yet to be established) is essential. The point is elaborated later.

Fig. II.1 is based on the assumptions of: complete combustion; inlet conditions of 60°F; and that the air required per Btu is the same for all fuels. Changes in specific heat with temperature have been taken into account. The use of the graph may be in error to a greater or lesser degree on account of three main factors, as follows:

(1) Inaccuracies due to averaging of all fuels. These are unlikely to be important; however, if greater accuracy in calculation is required for a specific fuel, the Rosin-Fehling It - diagrams and tables may be used. They are given conveniently in "Technical Data on Fuel" (Editor: H. M. Spiers) 6th. Ed., 1961 pp. 98-109. (World Power Conference: British National Comm. London).

(2) Inaccuracies due to incomplete combustion. If all combustion losses (both grate and/or stack) are known or estimated, the completeness of combustion fraction, ϕ, must be used as a multiplier for the available heat or nominal thermal efficiency obtained from Fig. II.1. Thus a nominal efficiency of 50% at a completeness of combustion of 95% means a real efficiency of 47.5%.

(3) Inaccuracies due to inlet conditions differing significantly from 60°F. This discussed below.

II.5 Influence of Inlet Conditions - In arctic conditions the ambient temperature will commonly be 50 to 100°F below the engineering normal of 60°F (used as base for Fig. II.1). This can affect the heating requirements in two ways. First it reduces the thermal efficiency for a given flue gas temperature, and second it can affect the heat demand (Sec. 1.3). These factors may be estimated as corrections to the principal calculations of Fig. II.1.

(1) Thermal Efficiency Influence - This is most conveniently estimated by calculating the additional heat required to heat the incoming air, from its inlet temperature to the standard 60°F, as a percentage of the gross heat developed in the heater. For one pound of fuel of calorific value B (Btu/lb), the stoichiometric air requirement is (B/100) (1+E). If we assume inlet at minus 40°F, a specific heat (average) of 0.25, and a cold air density 0.07 lb/cu.ft. at the reference temperature of 60°F, the heat required to raise the air from inlet to standard 60°F (a rise of 100°F) is

$$0.07 \times (B/100)\ (1+E) \times 0.25 \times 100 \text{ Btu/lb.}$$

This is the heat requirement per B Btu dissipated; so the percentage per unit Btu dissipated requires division of the above expression by (B/100). Taking E as 0.5, we obtain:
Additional thermal load due to heating inlet air from -40°F to +60°F

= 2.5%.

This is therefore roughly equivalent to reducing the thermal efficiency by about 1% per 40 deg. drop below the standard 60°F. This is clearly a factor that will generally be minor in comparison with the effects of changing excess air. Excess air is usually unknown and uncontrolled, but it can easily swing either way by 30 or 40% in many natural draft installations since it is very dependent on draft and therefore on the combustor performance itself. From this latter point of view, of control by draft, the low outside air temperature might have some advantage if draft delivers volume rather than mass since the air density will be higher at the lower temperature so that a given mass flow can be obtained for a lower draft. (This is behavior that also improves the efficiency of jet engines.)

(2) Heat Demand Influence - If the air is supplied to the combustor directly from the outside (drawn by draft) the efficiency penalty will be 1 to 3%. This is represented as a fraction of the heat that is unavailable to satisfy the heat demand. If the combustion air is drawn from the heated enclosure itself, the efficiency penalty is largely eliminated, but the heat demand is then increased. We have estimated that 1/3 cu.ft. of air is replaced every hour (Sec. II.2). If the replaced air is heated by 80° to 100°F from an outside temperature in the range of minus 60 to minus 40°F (i.e. assuming a "viable" temperature in the enclosure around 40°F), a similar calculation to that given above yields an additional heat demand of 0.5 Btu/hr.cu.ft. of heated space. If the actual heat demand is about 10 Btu/cu.ft.hr. (i.e. 20 Btu/cu.ft.hr. dissipated at 50% efficiency), the extra heat demand is equal to about 5% per 100 deg. temperature difference, or 1% per 20 deg. Subtracting the efficiency penalty of 0.5% per 20 deg. the net loss is about 0.5% per 20 deg., which is the same as for case (1) above and once again is relatively unimportant.

There is therefore no essential difference in thermal efficiency whether the air is drawn directly from outside, or whether it is drawn, already partly heated, from the enclosure; and this result is only partly coincidental. The heat for raising the air temperature must still come from the combustion and if the preheating of the air is complete in either case before the air enters the fire, the system does not "know" whether the air was drawn from the enclosure, or from outside with heating on the way. In that event, the drop in efficiency with falling outside temperature must be the same in either case. If, however, the air drawn directly from outside is heated in the flame itself, the flame and gas temperatures will be proportionately lower all through the system, thus reducing the rates of heat transfer and thus the heating efficiency.

(3) Other factors - The argument in favor of drawing the air from the enclosure is, therefore, that it may be (marginally) more efficient on account of slightly higher flame and gas temperatures. Arguments against this practice are: (i) that the preheating to preserve flame temperatures can still be achieved by drawing the air directly from outside through a simple heat exchanger; (ii) that the air coming in to replace the combustion air creates drafts; (iii) that the draft then becomes very erratic and is likely to be uncontrollable; and (iv) that the inleaking air is likely to overcool part of the enclosure fabric and increase losses (and thus the heat demand) in consequence.

Altogether, however, the potential for reducing efficiency is relatively marginal compared with such factors as excessive traffic in and out of the enclosure with a hot bubble of air escaping each time (unless precautions are taken). Another point to consider is whether some greater rate of replacement of air than once every three hours is not, in fact, desirable for physiological reasons. Although determination of this requirement is not within the scope of this report, and is one to be considered by a competent authority, nevertheless data on air changes given by Himus (loc. cit.) are possibly relevant. Desirable number of changes are given as two to three per hour. That is six to nine times more than the above estimates based solely on combustion needs. If two or three changes per hour is also a realistic value for an arctic hut or tent (due to the natural traffic in and out) then this rate of replacement of air can account for as much as 1/3 to 1/2 of the heat demand when temperatures are very low (say minus 40 to minus 60°F). This illustrates a significant point that the necessary information on the make up of the heat demand would seem to be missing at present. It may already exist in the literature although a brief survey has not turned up anything useful. The information is not immediately essential for this present report, but it could be of value later in optimizing the heating system in relation to the heat demand.

II.6 Other Quantities: Approximate Weights and Dimensions - From the required air rates it is now possible to estimate some other quantities such as the firing rates, required combustion volumes, stack dimensions, and so forth.

II.6.1 Firing Rates - As a matter of practicalities, the use of gas in the bunker heaters is not envisioned. We need be concerned only with liquid and solid fuels, and these can be taken as having calorific values lying between 5,000 to 20,000 Btu/lb. Typical liquid hydrocarbon fuels have values in the region of 18,000 Btu/lb. A good quality coal can be taken as 12,000 Btu/lb. Refuse with a fair proportion of paper is about 5,000 Btu/lb. (Somewhat wet.) Therefore, a small tent with a gross heat requirement of 10,000 Btu/hr. would require a little over 1/2 lb. of oil per hour, or a little less than 1 lb. coal/hr., or about 2 lbs. solid waste per hour. The larger bunker or hut with a gross heat requirement of 100,000 Btu/hr. would require roughly 10 times as much of each of the three fuels considered.

II.6.2 Combustion Volumes - These can be estimated from combustion intensities. Typical values of combustion intensity for natural draft firing systems are in the region of 20,000 Btu/hr. cu. ft. of combustion space. Combustion capacities of 10,000 Btu/hr. (Sec. I.1) therefore require 1/2 to 5 cu.ft. of combustion space. These volumes correspond to cylinders with diameters equal to their height with diameters roughly of 10" and 21" respectively. It should therefore be possible to satisfy the specified requirements for heat release rates by units of diameters not much more than one foot and two foot, respectively, with about the same height, or a little more to allow space for an ash pit when burning solid fuels.

II.6.3 Flue Pipe or Stack Diameter - This can be calculated from the air rates if we use the standard rule (empirical) that the flue gas velocity should not exceed 15 ft/sec. at the stack temperature. We have that the air flow rates (approximated as the gas flow rates) are 150 to 1,500 c.f.h. for the extreme limits of heating capacity of 10,000 and 100,000 Btu/hr., taken at 50% excess air (Sec. II.3). We do not know the stack temperature, but a value of 1400°F (which is high) has been suggested (Sec. II.4) for a thermal efficiency of 50% at 50% excess air. This temperature will therefore be used. The volume increase on heating is therefore 3.5 to 4. To allow a margin for moisture and increased volume on combustion, the volume increase factor of 4 will be used. The gas volumes to be handled therefore range from 600 to 6,000 cfh or 10 to 100 cfm. The required stack areas to accommodate these volumes are therefore 1.6 to 16 sq. ins; and the corresponding diameters are about 1.5" and 4.5". The upper limit value is realistic although the practical sizes are more often 3" to 4", indicating generally higher excess airs with lower temperatures and slightly higher velocities (up to 20 fps or a little more). In general, the higher velocities are to be avoided (unless needed for heat transfer purposes) since they represent loss of draft by extra friction. This is always to be avoided if possible.

The stack diameter of 1.5 in. at the lower rating, however, is likely to prove a little on the small side. A 2" to 2.5" dia. pipe would be appropriate. This follows from estimates of the Reynolds Number. For the

1.5" dia. pipe, Re is about 1300. By increasing the pipe diameter to 2" or 2.5" Re increases to about 2000 and the friction factor, which affects the draft losses, almost halves. By comparison, the 3" or 4" diameter pipe has a value of Re in the range 3000 to 4000. Since the flow is in the transition region between laminar and turbulent, the friction factor is rising with Re, but only by about 10% per 1000 increase in Re.

It should be noted that the stack diameter will not vary very greatly (say from 2" to 4") for the extreme limits of heat release (10,000 to 100,000 Btu/hr). Neither should the stack diameter need to be changed to any degree if the excess air increases, and the temperature drops to correspond, because the product of the excess air and the stack gas temperature is almost a constant under such conditions.

II.7 Stack Draft - If a new heater design is to be any improvement a key factor will be effective mixing between air and fuel, which depends on jet penetration. This is governed by jet velocity and ultimately by draft. Good mixing is especially critical when burning any fuels with a tendency to produce smoke. The combustion mechanism aspects are elaborated in Part III. Here we are concerned with generation of draft. There are two aspects here: first, what draft is required; and, second, what draft is possible.

II.7.1 Draft Required - There is almost impossibly little information available on the draft requirements. In "The Efficient Use of Fuel" (H.M.S.O. London 2nd Ed. 1958) pp. 130 and 261 some data are given for burning coal with drafts of the order of 0.15 to 0.5 in. w.c. The bunker heaters contemplated will almost certainly have to operate with drafts at the lower end of this range, and possibly below 0.1 in. w.c. An alternative estimate is available from a recent paper: "Development of Fundamental Basis for Incinerator Design Equations and Standards" by R. H. Essenhigh and Ta-jin Kuo. (Published in the Proceedings of the Third Mid-Atlantic Industrial Wastes Conference, University of Maryland, 1970, pp. 105-146). In Fig. 4 of that paper, collected data relating combustion intensity (I) to relative pressure drop ($\Delta p/p$) show that I is approximately proportional to $(\Delta p/p)^{1/2}$. The relation can be written

$$I = K(\Delta p/p)^{1/2} \qquad \text{(II.1)}$$

where $K = 1.5 \times 10^6$. However, K can be as high as 3×10^6 and as low as 1.0×10^6, with the range due, apparently, to the physical dimensions of the system. K generally decreases with increasing physical size. Since 0.1" w.c. is a relative pressure drop of 0.00026, the expected combustion intensity is 25,000 but could be as high as 50,000. At half that draft, or 0.00013 relative drop, the combustion intensity drops to about 18,000 but could be as high as 36,000. This means that drafts of 0.05 to 0.1 in. w.c. (net) may bracket the combustion intensity of 20,000 assumed earlier in estimating the combustion volumes (Sec. II 6.2). However, this leaves very little margin for pressure losses due to friction, turning corners, etc., so intensities may have to lie in the range 15,000 to 20,000 Btu/cu.ft.hr. This would double the combustion space required, but the linear dimensions would only have to increase by about 25%. The estimated diameters would therefore increase to just over one foot and two foot respectively. A combustion intensity of 10,000 Btu/cu.ft.hr. should be possible at a draft of about 0.02 in. w.c. - if the extropolation of the collected data is justifiable and applicable.

II.7.2 Draft Available - This can be calculated by a standard equation given, for example, in "Efficient Use of Fuel" (loc. cit.). If H is the stack height (ft.) and D is the draft in inches w.c., then for R_1 and R_2 as the temperatures in degrees Rankine for the ambient air and mean of the stack gas, respectively, then

$$D = H((7.6/R_1) - (7.9/R_2)) \qquad \text{(II.2)}$$

This formulation takes into account the slight average differences between the specific gravities of the air and combustion products.

In Fig. II.2, D is shown for the range of stack temperatures from 400 to 1400°F, at 4 different values of ambient temperature: 60°F, 32°F 0°F, and -40°F; and for a stack height of 10 feet. This stack height was chosen as being realistic for most installations. It may be taller in the larger huts, but every effort should be made to maintain the 10 foot height as a minimum.

The graph shows that, if application of eq. (II.1) is valid, then a combustion intensity of 20,000 Btu/cu.ft.hr. is possible for stack temperatures greater than 600°F and ambient temperatures below 60°F, but so long as the margin required for additional losses is small. If draft losses become appreciable, the requirements are aided by a lower ambient temperature, but if this provides insufficient margin, the margin can only be increased by increasing the stack temperature (which reduces thermal efficiency), or increasing the stack height, or designing for a lower combustion intensity. However, the reaction temperature will generally drop as combustion intensity drops, thus reducing the rate of heat transfer from the flame to the stove body so efficiency may drop. At these low combustion intensities there may also be some problems of flame stability and an increased tendency to smoke.

II.7.3 Other Factors - (1) The above assessment neglects the possibility of gaining additional draft in special cases by harnessing the stagnation effect of a wind. It would require only some sort of venturi throat at the top of the stack to increase suction, and/or a pipe with a bend pointing in the wind direction to develop some degree of forced draft. This is neglected at present on two counts: (i) It represents an easier combustion condition, and for the time being the focus should be on the most difficult condition which is natural draft without any aids; and (ii) it is a situation best taken into account after a primary design has been assessed. Indeed, in a high wind the draft and hence the combustion developed could be so intense that the heater might be damaged by overheating. It is more likely that controls to reduce the draft will be necessary. However, since the heat demand will also increase with wind velocity, the same factor will permit increased burning rates to match. To provide the necessary match between increased heat demand and dissipation, more information is first required on the constituents of the heat demand and their relative magnitudes.

(2) A final factor now to be considered is the feasibility of some of the conclusions developed. In particular, a flue gas temperature of 1400°F has been used for initial calculating so that the expected efficiency will more likely be on the conservative side. However, a temperature of 1400°F (or 760°C) represents red heat for solids at that temperature. There is no doubt that the common metals of construction would not last very long if they are continually heated or maintained at that temperature. On the other hand, Fig. II.2 (if valid) would indicate that the temperatures should not be allowed to drop below 800 or 900°F (430 to 480°C) if 20,000 Btu/cfh combustion intensity is to be attained. There is also the problem of how to drop the temperature. This can always be done to any required degree by increasing the excess air (Fig. II.1). If the same thermal efficiency is to be achieved, the excess air would have to be about 150%. The question is then whether the same efficiency could be achieved, and this depends on the heat transfer characteristics between the flame and the walls of the heater unit. An alternative would be to try to cool the gases before entering the stack, or in the stack, by means of a simple heat exchanger. This needs more elaborate discussion, but one method could be to construct the stack of two coaxial tubes, with the inner one carrying the hot gases, and the annulus between the two to carry the downward flowing cold air. Precautions would then have to be taken to make sure that the outer tube was never opened to the space being heated since it would then act merely as an additional way of escape for the heated air in the enclosure.

II.8 Summary - Approximate dimensions and magnitudes of a number of factors can be established that will be constraints to any design whatever the fuel or design-effectiveness for procuring good combustion. Some of the parameters do depend on the size of the enclosure (or heater output) to be heated even at the same specific dissipation rate or specific heat demand; other factors do not depend on the enclosure or heater size.

(1) For heaters in the capacity range 10,000 Btu/hr. to 100,000 Btu/hr., operation at 50% excess air and 50% thermal efficiency would seem to be realistic.

(2) The air rate for such heaters therefore ranges from 150 to 1500 cfh., approximately; and if the air is drawn from the enclosure being heated, this corresponds to one air change about every three hours.

However, it is not known whether this rate of air changing also corresponds to physiological requirements.

(3) Combustion intensities of 15,000 to 20,000 Btu/cu.ft.hr. would seem to be realistic and attainable. Combustion volumes for the two extremes of heat dissipation are therefore of the order of 1/2 cu.ft. and 5 cu.ft. respectively. These volumes could be accommodated in cylinders of height equal to diameter with diameters, roughly, of one ft. and two ft. respectively for the 10,000 and 100,000 Btu/hr rates of heat dissipation.

(4) The weight of fuel burned per hour will depend on the nature of the fuel. With oil at nearly 20,000 Btu/lb at one extreme and waste at 5,000 Btu/lb at the other, the rates of burning to satisfy 10,000 Btu/hr would be 1/2 lb oil per hour and 2 lb waste per hour. To satisfy 100,000 Btu/hr, the weights of fuel increase by a factor of 10.

(5) The stack diameters to handle the exhaust gases from the heaters at the two extreme limits should only be 2 in. dia. and 4 in. dia. respectively.

(6) Thermal efficiency of the heaters will be affected by the ambient temperature; but the extra load due to heating colder air will only correspond to something less than 2.5% drop in efficiency in most circumstances.

(7) At 50% excess air and 50% thermal efficiency the stack temperature would be 1400°F which creates a materials problem. Reducing the stack temperature can create a draft problem.

(8) A realistic stack height would be 10 ft.

(9) To attain a combustion intensity of 15,000 to 20,000 Btu/cu.ft.hr., with a stack height of 10 ft., the draft must exceed 0.07" or 0.08" w.c., and this in turn requires stack temperatures to exceed 700 or 800°F.

(10) The above values are subject to revision if better data (particularly on draft and combustion intensity) show significant inaccuracies in the assumptions made. However, it is not to be expected that the values will be changed appreciably, with the possible exception of draft requirements. These are, in fact, about the most critical of all the varied design factors and they are, unfortunately, the requirements for which the existing data are least accurate. This point should be well remembered in all that follows.

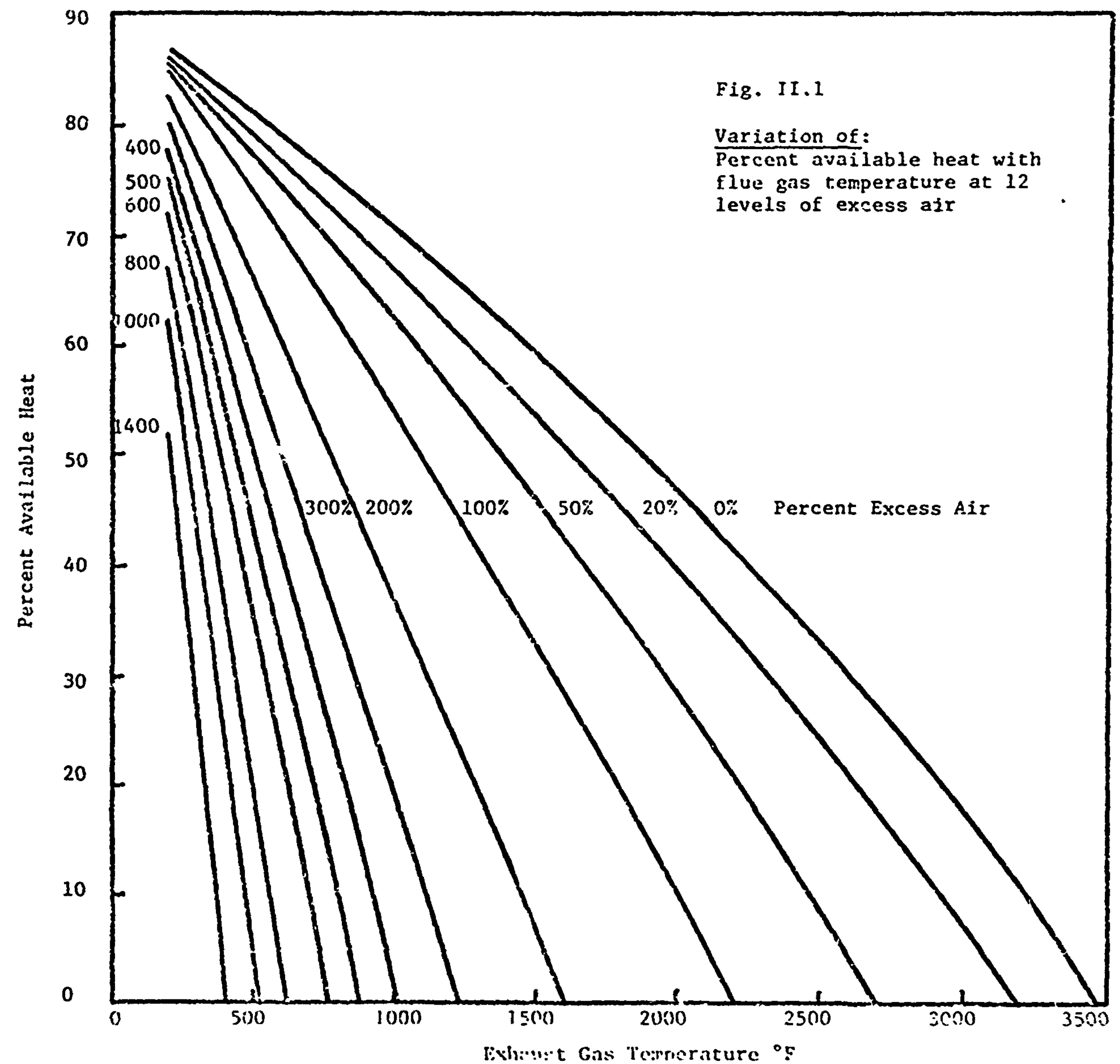

Fig. II.1

Variation of:
Percent available heat with flue gas temperature at 12 levels of excess air

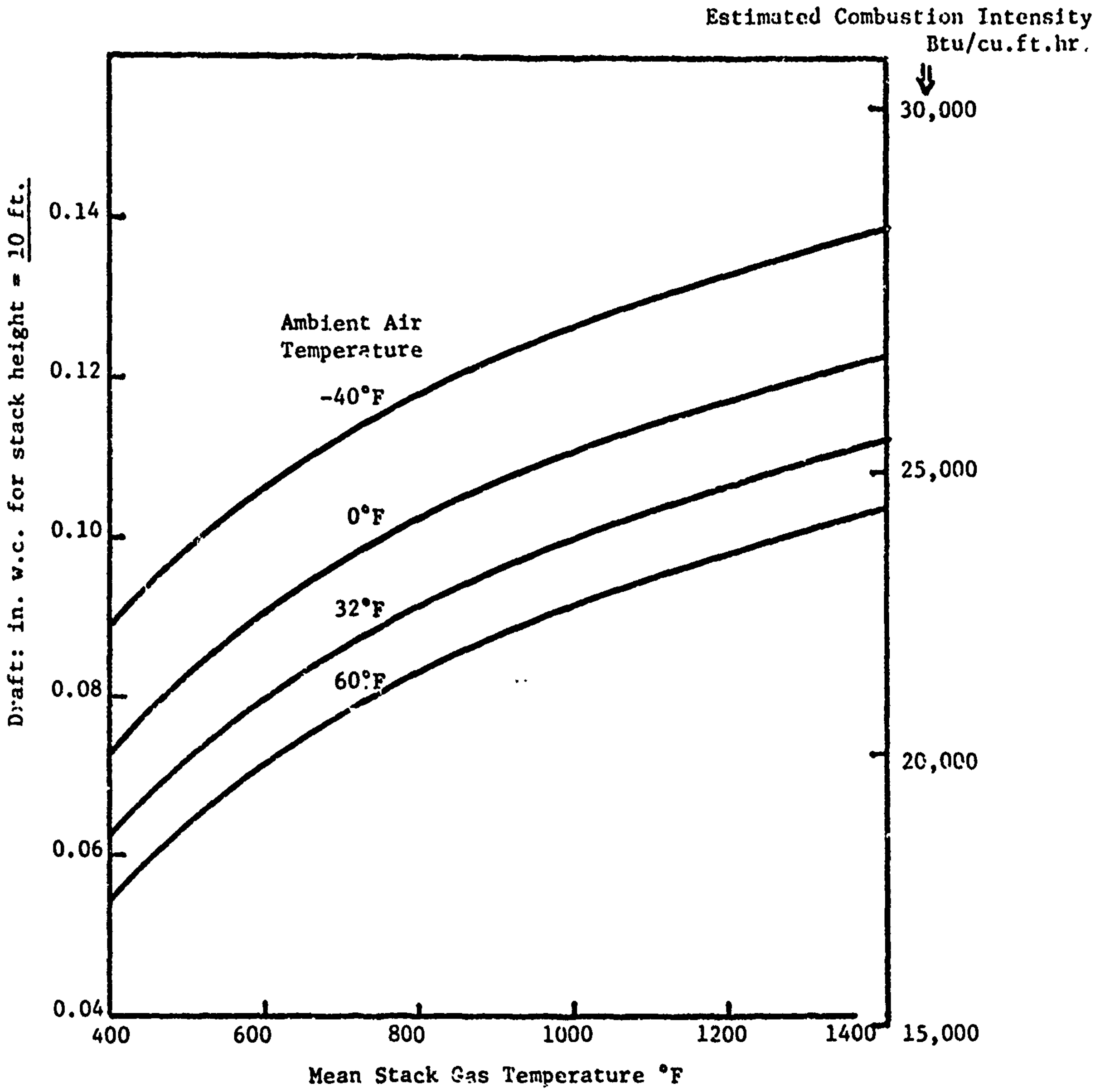

Fig. II.2 Variation of Stack Draft with mean Stack Gas temperature (in °F) for four different Ambient Air temperatures. For a stack Height of 10 ft. The draft is directly proportional to the stack height.

The Combustion Intensities are calculated from eq. II.1: Note - The validity of extropolation and application to this region has yet to be justified.

PART III: GENERAL MECHANISMS OF COMBUSTION AND HEAT TRANSFER

III.1 Introduction - The matter now to be considered is the general process of combustion including heat transfer inside the combustion chamber of the bunker heater. This section outlines some of the problems resulting from the combustion process. Possible solutions are discussed in sections following. We assume that fuel of one sort or another is fed into the combustion space, along with an appropriate (or inappropriate) quantity of air. We have to consider now

(1) The circumstances under which combustion will or will not be complete - determining combustion efficiency.

(2) The mechanism of heat transfer and heat recovery from the flame and hot gases - determining thermal efficiency.

Detailed mechanisms - referring to factors such as chain reaction processes, etc: - will not be discussed as they are either out of place or irrelevant within the context of this appraisal.

III.2 Fuels and Their Combustion Phases - The fuels of concern are only expected to be solids or liquids: as a matter of practicalities the use of gas in the bunker heaters is not envisioned (see Sec. II.6.1). The actual nature of the fuels is expected to range from good quality fuel oil, coal or wood to crankcase oil, refuse, or any other available solid or liquid hydrocarbon. On heating, such fuels will do only one of three things.

(1) Totally evaporate (light oils) or pyrolyse (some waxes and plastics) to form a combustible vapor only.

(2) Partially pyrolyse to form a combustible vapor but leaving a solid residue mainly of carbon (coals, woods, some heavy oils).

(3) Heat up without any change of phase or pyrolysis (this behavior is confined essentially to coke or char).

Again as a matter of practicalities we can virtually rule out a non-pyrolysable solid such as coke. This is generally more expensive than the coal from which it was derived and is more difficult to burn. Unless coke is produced locally as a by-product of towns gas manufacture it is not readily available, and it does not make sense to transport it into another area as a fuel, in preference to cheaper, denser coal, except in the interests of air pollution abatement.

Nevertheless, any closed stove capable of burning coal will generally be able to burn coke of normal reactivity. However, the basic design must be aimed at fuels that burn either

(1) In the gas, vapor, or droplet phase entirely

or (2) In the gas/vapor phase with a char burning in the solid phase.

The element common to both situations is the factor of combustion in the gas/vapor phase. As a convenience, the phrase "gas-phase" combustion is used hereafter to mean combustion of a gas, or a vapor (e.g. from an evaporating oil). Oils can also be burned as a droplet spray, this being a standard technique for domestic and industrial oil burners. However, generation of droplet sprays requires the use of oil pumps (for pressure atomization) or air pumps (for air atomization) so, again as a matter of practicalities, spray combustion is ruled out because of the absence of appropriate electric power facilities to drive pumps in most circumstances relevant to the use of bunker heaters.

III.3 <u>Production of Gas and Vapor</u> - The means by which the available fuels generate gas or vapor is simply the result of heating, but the circumstances under which the heating is arranged to take place can be usefully divided into the following:

(1) Vaproization in the feed tube before injection: this is accomplished, for example, in the kerosine blow torch or the primus stove, and it requires some slight pressurization to overcome the back pressure created on vaporization in the feed tube.

(2) Vaporization or pyrolysis inside the combustion chamber. This is the only conceivable method for burning charring or coking materials such as coal or wood or the heavier oils. It can also be used just as satisfactorily for the kerosines or other fuels capable of being vaporized in the feed tube.

The feed-tube-vaporizing type of system has the significant disadvantage that its use is restricted to fairly clean, light oils that do not easily crack (form carbon) on heating. However, even kerosines can be overheated if the stove is not properly operated, and the resulting carbon formation can clog the feed tubes, causing loss of flame. Brame and King in "Fuel" (p. 333) indicate that distillation cuts below 300°C are required to avoid clogging by carbon formation under normal operation.

The heat required for vaporization or pyrolysis is, of course, derived from the subsequent combustion of the pyrolysed fuel.

III.4 <u>Requirements for Satisfactory Gas-Phase Combustion</u> - The situation therefore envisaged is that of a vapor projected either from a feed pipe bathed by the flame (primus type system) or from liquid flowing into a trough or up a wick, or from pyrolysing or evaporating solids on a grate. For this vapor to be totally burned without emitting smoke a number of conditions have to be satisfied:

either (A) With care, the more volatile liquid fuels can be burned off wicks, but establishment of no-smoke conditions is often difficult

or(B)(1) Air has to be injected into the vapor in such a way that as uniform a mixture as possible will be formed.

(2) The vapor-air mixture has either to be simultaneously mixed with already-burning gases or it has to be injected onto a red hot ignition refractory. This is the process of flame stabilization.

(3) The still-burning gases must then flow away from the flame holding region whilst maintaining sufficient temperature for complete burn-out.

(4) Throughout the whole of the mixing and combustion process, no significant fraction of the fuel must be allowed to mix with insufficient air and then to heat up since this causes smoke. Unfortunately, as the exact mechanism of smoke formation is still obscure, the exact conditions to be avoided are also unknown. In general, however, prevention of smoke depends on good mixing and sufficient oxygen.

(i) The most critical factor is flame stabilization and here we can refer to the results of jet engine and rocket research of the last 20 years as an aid to requirements for burning, except off wicks. According to this past research, ignition and flame stabilization may depend critically on generating as uniform a mixing region as possible where the fuel and air first enter. This is known as the "stirred reactor" concept. A critical component of such a mixing pattern is now known to be a "backmix" flow by which convective streams of hot gas and part-burned combustion products move upstream against the overall direction of flow. There is also evidence that a strong backmix flow is crucial in preventing smoke formation (to judge by evidence from jet engine combustor cans) or conversely that a weak backmix flow will permit smoke formation. Experiments by the writer on burning waste (card) where volatiles or liquid particulate smoke was the fuel (as in coal or wood burning) demonstrated that without adequate mixing, some smoke would pass right through the flame and escape even in the presence of 50% excess air, whereas the smoke could be easily burned up by adequate mixing. At issue, in consequence, is the means of achieving the necessary mixing as discussed below.

(ii) The essential need for the "stirred reactor" region for flame stabilization depends on the throughput velocities for, if the injection velocity or flow velocity exceeds the flame speed, the flame will be lost by blow off. On this basis the flames in bunker heaters should, in fact, stabilize without need of a stirred reactor section. This conclusion follows from the estimates of flow velocities in the bunker heaters. According to Part II of this Report, the air requirements (cold) were estimated at 300 cu.ft. per hour per cu.ft. of combustion space. The cold residence time is therefore about 12 sec.; and the hot residence time for a flame temperature averaging 900°C, is 3 sec. In that time the distance travelled is one to two feet (30 to 60 cm.) giving average flow velocities of 10 to 20 cm/sec. Since laminar flame speeds (for premixed gases) are mostly in the range 20 to 100 cm/sec., the flame stabilization by velocity balancing should present no particular difficulty. This therefore permits stabilization of diffusion flames of the type formed on wicks (which usually have a leading anchoring point that is effectively premixed).

Oil diffusion flames, however, are famous for their smoking tendencies, particularly if the fuel is non-paraffinic and especially if it is aromatic. As the burning rate is increased, a point is always reached at which the flame starts to smoke at the flame tip even when using the favorable paraffinic fuels.

Consequently, although creation of a stirred reactor region would not appear to be an absolute essential from the point of view of flame holding, it would nevertheless seem to be highly desirable - particularly for the non-paraffinic vapors - if a serious smoking of the flame is to be avoided at some firing condition. Smoking is undesirable on two counts in this context: it represents reduced combustion and thermal efficiency, and it is militarily unacceptable when it results in a visible plume from the stack.

(iii) Associated with all flame holding devices, there must also be adequate space for final burnout. Lack of such space results either in combustion being completed in the stove pipe or stack, which overheats sections of the unit not designed for such temperatures, or else the gases entering the stove pipe are so far cooled that reaction is quenched and the combustible loss is thereby increased, with a corresponding loss of efficiency. The need for combustion space for final burnout should, however, be taken into account automatically in the overall estimates of combustion volume requirements from the combustion intensity assumptions (Sec. II.6.2). The interesting point is that the best information suggests that the magnitude of final unburned emissions often depends quite critically on the conditions in the initial reaction region, not in the burnout region.

III.5 Requirements for Satisfactory Solid-Phase Combustion - Only the char or coke-forming fuels (coal, wood, heavy oils) leave a solid combustible residue. Such a residue is mainly carbon, with some hydrogen; and inerts such as moisture and ash. Industrial fuel beds of coal or coke (on grates) are usually so deep that air flowing from underneath through the grate bars first burns off sufficient carbon to form carbon dioxide in the lower section of the bed, and this gasifies the higher section to form carbon monoxide as a major constituent of the gases flowing into the overbed region. If the unit operates on natural draft, most of the draft has to be applied across the bed where most of the resistance to flow occurs; and traditionally such natural draft units usually had little overfire air. To avoid problems with smoke emission then generally meant that the fuel used would be a hard "steam" coal or coke. With forced-draft, underfire air this problem was eliminated.

In the case of small household units, the location of the reaction zone can vary appreciably with design. With open fires, either on grates or on solid hearths, the air mostly flows in over the top of the fuel bed and then diffuses down into the reaction zone. With closed stoves it is more customary to arrange for the air to flow through the fuel bed - if this is not done, much of the value of the enclosure is otherwise lost. In either case, however, the burning particles of solid fuel must remain reasonably coherent and reasonably porous if the solid is to burn out. If the fuel particles collapse, the resistance to flow of air through the bed becomes too great because the bed is then too dense, and an adequate heat release rate to keep the bed burning cannot be maintained. At the same time, however, the particles themselves must develop substantial internal surface to provide necessary contact area with the indiffusing oxygen for, without that area, the reaction cannot be maintained at a sufficient rate, again to produce the heat necessary to keep the bed alight. Fortunately, this area requirement is a condition normally satisfied without difficulty by coals, woods, and some cokes. The reaction zone is usually small (or thin) and the temperatures are generally low enough for appreciably more carbon dioxide to be formed than is the case for deep beds.

A potential problem can be anticipated, however, when burning certain heavy fuel oils. Single particles of many fuel oils will burn in two stages, like coal particles, with vapor boiling off first, leaving a solid coke residue that burns as carbon. There can be difficulty in handling such fuels in quantity, however. If the oil is already fluid, or turns fluid on heating, it obviously cannot be burned on a grate, (the bunker oils are often so viscous that they won't flow at all at normal temperatures). The char formed then presents difficulties for burnout. The oil has to be burned in a trough so that, even if any reasonably coherent char is formed, it is difficult to design a system utilizing underfire air jets where the jets will not either be clogged, or so small that no appreciable quantities of air can ever be forced through them by natural draft. Furthermore, the problem is usually aggravated by the naturally dense char commonly formed from heavy oils and other similar materials such as bitumens or asphalts. Information on this behavior is critically important from the design point of view, but is largely missing so far as natural draft stoves are concerned.

III.6 Gas Phase Mixing Processes - Given an adequate flow of vapor from an evaporating or pyrolysing fuel, where wick stabilization of the flame is undesirable or impossible, the balance of information is in favor of creating the stirred reactor or well-mixed region mentioned in Sec. III.4. Such mixing may not be absolutely essential for flame stabilization, but the evidence seems to be that it is essential for adequate combustion efficiency and low smoke emissions. This mixing requirement is at the focus of the draft problem discussed in Part II, Sec. II.7. The incoming air has to promote adequate mixing between the air, the fuel vapor, and the combustion products already present; and part of the flow pattern that will achieve this must be a generated backmix flow: And the only source of energy to drive this backmixing process is from the combustion draft.

In Sec. II.7 the magnitudes of the draft requirements and draft available are discussed. If the available data can be validly extrapolated, that previous examination does indicate that sufficient draft for adequate mixing is available. What has to be considered now is the best means of applying that draft.

In general, it would seem that air jets directed radially across the fuel vapor flow will promote some degree of mixing with backmixing, but in general radially directed jets are disadvantageous. If a jet had only sufficient strength to penetrate to the center of the combustion tube before its energy is dissipated, then the natural paraxial flow is usually strong enough to deflect the jet flow downstream and prevent that jet from generating a backmix flow. A stronger jet that is not deflected, however, will also then have sufficient strength to cross the combustion chamber, or nearly so, before dissipating, and this can promote mixing near the walls at the expense of mixing at the center. Two opposed jets designed to meet at the center will achieve mixing at the center, but the resultant jet of the combination then turns to flow axially downstream, again at the expense of backmix. One solution appears to be to offset the jets to flow chordally, not radially, but all in the same rotational sense to generate a vortex. This seems to generate a much stronger mixing pattern. A further addition that appears to be also advantageous is a second set of jets generating another vortex further downstream, but rotating in the opposite sense. Both the single and the double vortex seem capable of producing satisfactory levels of backmix flow.

A third possibility, not tried but suggested for consideration is to arrange the air ports to produce jets flowing upstream in the first instance. If the jets are also arranged to scour a carbon or char surface, such jets might serve double duty to burn up any char residue from wood or a heavy fuel oil whilst at the same time generating the necessary backmix in the gas phase.

III.7 Other Vaporizing Burners - Easily volatilized oils will, of course, burn quite easily under a variety of conditions, such as in a cup or just poured out onto the ground. However, the copious smoke usually emitted is clearly indicative of low combustion efficiency. Some considerable improvement is achieved by the so-called "pot" burner often used for space heaters of 60,000 to 80,000 Btu/hr capacity. This has some marginal pretence to design, with a perforated co-axial liner inside the pot and a flame ring half way up. Vapors rise in the pot to mix with (primary) air flowing in through the holes in the liner. The device is sufficiently adequate to sustain a commercial market, but as Smith and Stinson report (p. 224 of "Fuels and Combustion", McGraw Hill, 1952) "control of the air and fuel is difficult, and somewhat less efficient combustion results along with larger formations of sludge and carbon than with other types of burners". The unit must be cleaned periodically.

All existing pot (vaporizing) burners with possibly one exception appear to suffer from these disadvantages. The exception is the Aladdin "Blue Flame" burner in which the simple perforated liner and flame ring is, in effect, replaced by a series of flame rings or plates (the unit is rectangular, not circular) with primary, secondary, and even tertiary air supplied in a staged manner through appropriately located perforations. The result is normally blue-flame, virtually odorless burning, but the logic of the arrangement that creates this end result is unknown.

The critical conclusions from this comparison of pot burners is that: (1) they will burn high-volatility oils quite easily with no particular flame holding problems (because of generally low fuel and air velocities); (2) but the devices are generally not too clean burning; and (3) improvements (e.g. the Aladdin burner) are possible; consequently (4) further improvements (e.g. imposition of backmixing) may also be possible, with clean burning of higher-carbon (lower volatility) fuels also possible.

III.8 Air and Fuel Control - A major problem integral with that of the combustion process is the problem of air and fuel control.

(i) Supposing first that the fuel supply rate is constant, the air supply rate can depend critically on external factors such as outside temperature and wind speed. If the wind is gusting, the air supply can swing from deficient to high excess and back again to deficient in a matter of minutes or even seconds. This can completely upset the heat dissipation rates and expected fuel consumption. What is clearly needed is a feedback device that will partly at least damp out the draft fluctuations. The traditional answer is an atmospheric damper at the bottom of the stack which opens in response to increased draft and short-circuits the air flow. This can be contemplated although there are two possible objections to it in the context of bunker heaters. First, the valves are commonly quite flimsy so they will respond quickly. For mobile field use the existing designs may be just too flimsy and liable to damage. However, something less flimsy may still respond with sufficient speed for the purposes considered. Second, the valve usually opens into the heated enclosure (i.e. the tent, so heated air is lost up the stack). In the case of a well sealed tent, with a separate pipe for the combustion air supply, the valve may open but be unable to "draw". The solution here should be to connect the exhaust stack to the air supply pipe through the damper so that part of the extra air drawn in will short-circuit the stove and exhaust directly into the stack. Since this will partly cool the stack it will simultaneously reduce the draft and help to correct the condition that led to the damper opening in the first place.

(ii) The assumption that the fuel supply rate is constant is also often invalidated. Circumstances frequently cause violent fluctuations in fuel supply, sometimes but not invariably matched by equal fluctuations in burning rate. The problem is frequently akin to filling a bath when the drain is open, and trying to keep the water level constant. One is attempting to match two independent variables; there are too many constraints for the number of degrees of freedom. There are four cases to consider.

(a) The fuel vaporizes as fast as it is fed to the stove, and it burns as fast as it vaporizes. This is essentially the case with the primus stove, the blow torch, and the wick lamp or heater - up to the smoke point. At the smoke point, the fuel is still vaporizing as fast as it is supplied, but the air supply rate no longer adequately matches the fuel supply rate. In the case of the wick unit, the rate of fuel supply is in essence determined by the rate of vaporization (which in turn is influenced by the wick height which helps to control the flame size). Even so, design and adjustment of such units to prevent odor so smoke is quite difficult. In the case of prevaporizing units in normal operation (such as the blow torch) the heat available for vaporization always exceeds the heat required, so that all the fuel supplied is vaporized and the surplus heat available for vaporizing more fuel goes instead into higher temperature of the vapor. However, such systems can be swamped if forced, and a fuel feed rate can be reached at which only part of the fuel is vaporized.

In the normal operating range, however, there is an available degree of freedom in the system (such as increased vapor temperature) that allows the burning rate to change with the fuel feed rate without serious problems.

(b) The pot vaporizer, on the other hand, is a good example of an overconstrained system. The burning rate is normally controlled by a valve in the feed pipe (with either a pump or gravity feed). If the fuel supply rate is increased there is no initial reason to expect the vaporization rate to increase, and the first result is that the fuel level in the pot increases. Ultimately this does have some effect, and the vaporization rate (and hence the burning rate) will increase somewhat, but the increase is evidently in proportion over only a small range of firing rates. Consequently, the device is very inflexible. The Aladdin stove, in contrast, has an extra degree of freedom built in by the simple but ingenious expedient of arranging the liquid fuel to flow into a Vee shaped trough. Consequently, when the fuel feed rate is increased and the liquid level increases also (as described) the surface area of the liquid pool also increases thus allowing a matching vaporization rate, with the liquid level as a dependent parameter. It is then necessary, of course, that the rest of the system is so designed that the burning rate will still match the vaporization rate over a reasonable range of fuel inputs.

(c) Solid fuels, by contrast, often present quite a different set of problems. Solid fuel stoves have two basic classifications. They are designed either (i) with fuel hoppers, or (ii) without fuel hoppers. The difference can be reasonably critical. Those without fuel hoppers are batch burning systems. Those with fuel hoppers are semicontinuous or continuous. In the batch burning systems, the pyrolysing fuel generally creates a strong "flush" of volatile products (that the stove then has to handle without causing smoke) as it is charged. The production of volatiles then decays more or less exponentially with time (according to some authorities) until only the carbon residue is left. Unless the damper settings are adjusted continuously throughout this period the excess air - controlling efficiency and draft - will be changing steadily from something small (say 10 to 30%) to something large (anything in the range of 100 to 300% depending on the actual nature of the fuel). This makes heating very erratic and somewhat unpredictable. There is also a tendency to feed too frequently and overcharge the unit, to maintain the period of flaming combustion of the volatiles. What then happens is that the char bed increases in depth, and so does the heat release rate. However, it has to be assumed that, between the high fire rate during volatiles combustion or the low fire rate during char combustion, if one is adequate for the needed heating requirements, then the other is either too high or too low. Generally, it means that the high fire level is too high, and since the unit will not be designed to handle such copious quantities of volatiles, it will smoke heavily instead.

There are several answers to this variable output problem. One is to use hard coal or coke so that the quantities and duration of pyrolysis is greatly reduced or eliminated. Another is to use a fuel hopper so arranged that as the fuel burns to ash, the ash skeleton collapses and allows fresh fuel to feed down from the hopper. Numerous patented variations of the hopper stove are available on the commercial market. They are mostly of cast iron with refractory linings for certain sections, and are very heavy.

(d) The fourth case to consider is that of the viscous, high carbon oil (or bitumen or pitch, etc:) that will not flow into a combustor chamber unless heated, but cannot be put on a grate because it will flow when heated. Such residual fuel oils and even more viscous tars are used as fuels in industry, but always with heaters and tracer lines to keep the temperature high enough for mobile flow and pumping requirements. The burners are usually steam atomizing burners handling tens or hundreds of gallons per hour, and the flames obtained are very hot. Such handling methods are quite inappropriate for small bunker heaters. For these fuels we are therefore left with an apparently unsolved problem so far as an appropriate burner is concerned. No small unit to handle such fuels is known. It is also possible to predict that such fuels will probably present major (but not necessarily unsurmountable difficulties) in burning them in small units without preheating or power devices, without making smoke.

III.9 Heat Transfer - A significant component of the total combustion mechanism in all relevant cases is the heat transfer from the flame (or reaction zone) to the fuel, for evaporation or pyrolysis. Without that heat transfer there is usually no flame. The heat transfer mechanisms, at the same time, operate to heat the shell of the stove, and hence the unit enclosure (tent or hut). The quantities of heat required for evaporation or pyrolysis are relatively small. Latent heats for evaporation of hydrocarbons range mostly from 100 to 500 Btu/lb., and this also seems to bracket most heats of pyrolysis (except for some plastics which can be as high at 1,000 Btu/lb.). Compared with heats of combustion of 5,000 to 20,000 Btu/lb. (Sec. II.6.1) this is 10% to less than 1% of the heat of combustion. It is therefore customary to take the vaporizing heat transfer somewhat for granted so that the exact mechanism transferring most of the heat is probably as much a matter of speculation as of fact.

III.9.1 Heat Transfer for Evaporation and Pyrolysis - For high-volatile, clear-burning fuels the probability is that the main process of heat transfer from the flame to the fuel must be by conduction and convection. For higher-carbon fuels on the other hand with quite luminous diffusion flames, there is considerable evidence from pool burning experiments that radiation then contributes significantly. Temperature measurements in the pool show the top millimeters or centimeters (depending on the fuel) are at a uniform temperature with boiling throughout this region. This is interpreted as heating in depth by radiation. However, the difference between the low carbon and high carbon fuels may be in fuel absorptivity rather than flame emissivity. The flames in such experiments are of considerable size (of one or more meters) so the path length through the flame is substantial, also of one or more meters, and flame emissivity depends on path length (as well as radiant particle or species density). Nevertheless, in stoves of the size of bunker heaters, the contribution of radiation is probably significant even though emissivities will be less than 0.5 and will probably be less than 0.2 or 0.1. Emissivities of nonluminous flames can be estimated from the Hottel curves.

Of the other two methods of heating, conduction can operate either directly through the gases or through the metal (or ceramic) walls of the heater. In the pot vaporizer, for example, the heat is most probably transferred down the walls of the pot itself. In more open pools, as in the Aladdin burner it is more likely to be convection of the hot gases above the fuel surface.

If the fuel chars or cokes, and the char is then burning, the char bed must generally be able to maintain ignition in the absence of an overbed flame. The main source of heat for pyrolysis of fresh fuel is therefore the char bed itself, and this can be either radiation or convection. A burning char bed is a substantial source of radiation, with emissivities close to unity. If fresh fuel is fed underneath the burning bed the reaction zone can generally burn down into the fresh fuel region with the necessary heat transferred, according to the best available evidence, by radiation alone. If the fresh fuel is supplied on top of the burning bed, the radiative heat transfer is supplemented by convection. This, incidentally, creates problems in the case of some stoves which are simply steel cylinders about three feet high and one foot in diameter. If such a stove is lit at the bottom and then filled with fuel to the top, an inordinate quantity of fuel will heat up, and be burned up, in a relatively short period of time. Effective designs for hopper stoves must prevent the hot gases convecting up into the storage hopper. Since total exclusion of heat is almost impossible, such units generally have to be fired on coke alone since otherwise appreciable pyrolysis will occur in the hopper. Hopper-stoves are not therefore likely to be suitable as bunker heaters where the fuel may be coal or wood or even refuse.

III.9.2 Heat Transfer with Respect to the Stove Shell - Of equal concern from the utility point of view is the heat transfer from the flame to the stove shell, and from the shell to the tent or enclosure being heated.

(i) Heat transfer from the flame to the shell for the most part is governed by the same general considerations that govern heat transfer to the vaporizing fuel; but there are obvious differences in detail. The magnitudes are quite obviously substantially different. The percentage of heat exchanged from the flame and hot gases must be nearer 50% than 1% to 10% (for evaporation or pyrolysis), or roughly an order of magnitude greater. Consequently, the mechanisms of heat transfer must be correspondingly more effective. The quantities involved can be estimated approximately. In Part I (Sec. I.1) the heat dissipation rates in the heaters were estimated to range from 10,000 Btu/hr. to 100,000 Btu/hr. In Part II (Sec. II.4) a thermal efficiency of 50% was considered to be realistic, so the net heat transferred to the stove shell and thence to the heated enclosure would be 5,000 to 50,000 Btu/hr. For stoves regarded as cylinders (Sec. II.6.2) of height equal to diameter and of about 1 ft. and 2 ft. diameter for the two extreme cases, the cylinder surface areas would be approximately 3 and 12 sq. ft. respectively. If heat loss is mainly from the vertical cylinder surfaces (the top surface may include the stack connection, etc:) the heat transfer per unit area of surface must range from 2,000 to 4,000 Btu/hr. sq. ft. (approximately). (The difference between the two values reflects in part the different surface to volume ratio of the two sizes of stove). The Reynolds Numbers for gas flow in the stoves range from about 12,000 to 25,000 for the two cases, and the convective heat transfer coefficients (which are very difficult to estimate for such conditions) are probably in the region of 1 and 2 Btu/sq.ft.hr. respectively. Consequently, average temperature differences of 1,000°F between the flame and stove shell would provide roughly 50% of the required net heat transfer. Temperature differences of 2,000°F would provide it all.

Radiation contributions are even more difficult to estimate because of the greater relative uncertainties in both gas temperatures and emissivities. However, if we assume an inside shell temperature of 500°F (see below), the reradiation (black body) loss is approximately 1,500 Btu/sq.ft.hr. (or a little less). A black flame 1,000 degrees higher radiates at about 25,000 Btu/sq.ft.hr., and at 0.1 emissivity this becomes 2,500 Btu/sq.ft.hr. The net transfer from the flame to the shell is between 1,000 and 2,000 Btu/hr. (allowing for contributions from other parts of the stove shell) so radiation at this level would also contribute roughly 50% of the required net heat transfer. However, a flame temperature of 2,000°F higher than the stove shell, at 0.1 flame emissivity, would provide nearly 12,000 Btu/sq.ft.hr. which is half to one order of magnitude greater than the balance required after the convective contribution has been subtracted out.

These estimates are very approximate and ignore such refinements as view factors, etc.; however, they match previous estimates sufficiently well, with one exception, that we may reasonably conclude: (i) that the stove shell should be in the region of 500°F, and the effective, average flame temperature (heat transfer temperature) should be in the region of 1,500°F; (ii) the net heat transferred from the flame to the stove shell is then roughly 50% by convection and 50% by radiation, with flame emissivities in the region of 0.1. The flame temperature estimate is astonishly close to measured values (1,400 to 1,600°F) found in (liquid particulate) smoke flames formed in 3 inch diameter tubes. (Biswas and Essenhigh: Paper 39f: A.I.Chem.E. 70th Nat. Meeting, 1971.)

The factor that is inconsistent with these estimates is the relation of exhaust gas temperature (assumed previously - Sec. II.4 - to be 1,400°F) and heat transfer temperature, now estimated as being in the region of 1,500°F. However, Hottel suggests as a rule of thumb (Hottel and Sarofim, Radiative Transfer, p.468) that the two temperatures consistently differ by only 200 to 300°F. If we change the gas exit temperature to 1,200 or 1,300°F (650 to 700°C) this reduced value would meet expectations discussed in Sec. II.7.3 point (2) concerning the likelihood of such a high temperature as 1,400°F. The reduction would correspond to an increase of excess air to about 75% which may also be more realistic in many circumstances.

(ii) Heat transfer from the stove shell to the surroundings is more easily handled. Fig. III.1 from Technical Data on Fuel (Spiers) p.68, shows the heat losses by convection and radiation from surfaces. The graph shows that radiative loss at 500°F (assuming the inside and outside shell temperatures are similar) is about 1,500 Btu/sq.ft.hr., and the convective loss is about 500 Btu/sq.ft.hr. The sum of the two matches the lower estimate of heat loss density, of 2,000 Btu/sq.ft.hr., for the smaller stove. For the larger stove, the wall temperature would have to be increased to about 700°F with correspondingly slight increases - or none at all if the flame emissivity is increased - in the interior average temperatures. However, at this point the uncertainties in these various estimates outweigh the magnitude of the changes required and further minor changes in the figures have little or no significance. Within even a 10% or 20% error, the agreement between the figures is surprisingly good.

These estimates are for a unit assumed to be reasonably compact, and well operated, with a reasonably uniform overbed flame. In practice, we may expect the values to depart from these. Firing will be somewhat erratic; excess air could be 100 to 200%; improper stoking could easily lead to smoking; and the heat transfer surface could be larger than estimated if the top of the stove and the stove pipe are sufficiently hot to contribute materially. However, the level of uncertainty suggests that an early step towards unit heater improvements should be determination of an accurate heat balance on existing devices at several operating levels.

III.10 Air Preheat - The estimates of the previous sections are still consistent with stack gas temperatures being in the region of 1,400°F. If estimates in Part II (See Summary: point 9) are correct, that draft requirements might possibly be met by stack gas temperatures of 700 or 800°F, then nearly half the sensible heat in the stack gas may be available for recovery. This would increase the unit efficiencies from 50% to 65% or 70%. The potential improvement is much larger than these figures superficially indicate. If efficiency increases from 50% to 67% (an increase of 17%), it means that only 3/4 pound of fuel is needed where 1 pound was needed previously, a drop of 25% in the fuel requirement. Alternatively, a given tonnage of fuel that will last N days at 50% use efficiency, will last (N/3) days or 33% longer at 67% efficiency. The logistic significance of such increases, particularly in difficult terrain or weather conditions, will be self-evident.

Air preheat is a possible means of increasing the system efficiency. If a heat exchanger can extract heat from the exhaust gases and preheat the combustion air, the heat recovered will enable the fuel rate to be reduced in proportion. The preheated air will also improve combustion efficiency.

There can also be other gains. If the draft is fluctuating, increased draft generally increases both the excess air and the burning rate. Excess air will decrease the gas temperature (thus reducing the draft - and the efficiency) but increased burning rate will increase the gas temperature again. Heat recovery will reduce the exhaust gas temperature, thus tending to reduce the draft and hence offset the effect of the draft fluctuations.

On the debit side is increased complexity, unless the heat exchanger is something as simple as two coaxial tubes, or two tubes back to back, and are therefore easily cleaned if necessary even though integral with each other. Other devices might include double walled stove shells although these would again represent unwanted complexity and weight, with additional complications of heat transfer to the enclosure, damage and blockage, and draft restrictions due to narrow passages. Nevertheless, the gains from improved efficiency could be so significant that some increase in cost, weight, and complexity would probably be justifiable although the exact trade-offs have yet to be calculated.

III.11 Heater Radiation - Another method of improving efficiency is by directly increasing the heat loss from the stove itself. What could be particularly appropriate here would be refashioning the flue passages inside the heater itself, before going to the stove pipe, to heat special sections of refractory metal, if possible to red heat. At only 1,000°F the radiative loss (according to Fig. III.1) is nearly 8,000 Btu/sq.ft.hr. and is three to four times the convective loss. Clearly, however, the flame gases must exceed the temperature of the radiative panel, so further recovery by a heat exchanger might still be desirable unless the gases adjacent to the radiant panel are still burning.

III.12 Summary - Some limitations on fuels, firing methods, combustion requirements, and heat transfer rates can be established.

(1) Only evaporating liquids (light oils) and pyrolysing solids and liquids (coal, coke, refuse, heavy oils, tars) need be considered as potential fuels. Considerations of practicality and availability eliminate gases and nonpyrolysing solids such as coke.

(2) Devices therefore need to be able to handle fuels burning only in the gas (vapor) phase alone or in the gas and solid (char) phase simultaneously. The fuel may be prevaporized in limiting cases only (light oils); all other fuels must be vaporized or pyrolysed in the combustion chamber itself.

(3) Existing devices on the commercial market provide more or less adequate conditions for burning free-flowing vaporizing liquids, and pyrolysable solids, but not burning viscous pyrolysable liquids or liquifyable solids. Where combustion conditions are adequate they are still mostly somewhat limited.

(4) Improvements, to overbed combustion particularly, are believed to be possible with careful attention to overbed mixing, if natural draft can supply sufficient energy to air jets to promote backmixing. To improve mixing, three arrangements are suggested for initial examination: (i) the single vortex; (ii) the double vortex; and (iii) back flow jets.

(5) Attention to air and fuel control is also required. These are two interconnected but still separate problems. Fluctuations in firing rate can alter the draft and hence the air supply; and to a lesser extent vice versa (particularly pertinent to solid char combustion). Draft control can be the more important. Atmospheric dampers and cross-links between the stove pipe and an air supply pipe are possible means of damping fluctuations.

(6) Maintenance of fuel supply by pyrolysis or evaporation requires 1% to 10% of the heat of combustion to be fed back to the incoming fuel. This generally presents little problem with both radiation and convection contributing. Radiation is particularly important for flame spread through solids.

(7) Convection and radiation likewise transfer the heat from the flame to the stove shell, and thence to the enclosure being heated. The processes are consistent with the following estimates: that about 50% is transferred

by convection and 50% by radiation from the flame (at emissivity in the region of 0.1) to the stove shell. Heat transfer coefficients for convection are 1 to 2 Btu/sq.ft.hr. and the thermal flux densities are in the range 2,000 to 4,000 Btu/sq.ft.hr., with the latter value probably on the high side and the former closer to an average. Flame temperatures for heat transfer average 1,500°F, and stove shell temperatures are about 500°F. The heat transferred through the shell is then transferred to the local surroundings by convection, responsible for about 25%, and radiation responsible for about 75%. Even so, provision of special radiation panels of refractory metal should be considered to increase the radiative transfer still further.

(8) There is probably still latitude for increasing the overall efficiency - if the approximate calculations can be trusted by heat recovery in a simple heat exchanger. If the flue gas temperature can be reduced from 1,400°F to 700 or 800°F it corresponds roughly to an increase of efficiency from about 1/2 to about 2/3. Such an increase would correspond to an effective increase in the fuel supply by 1/3.

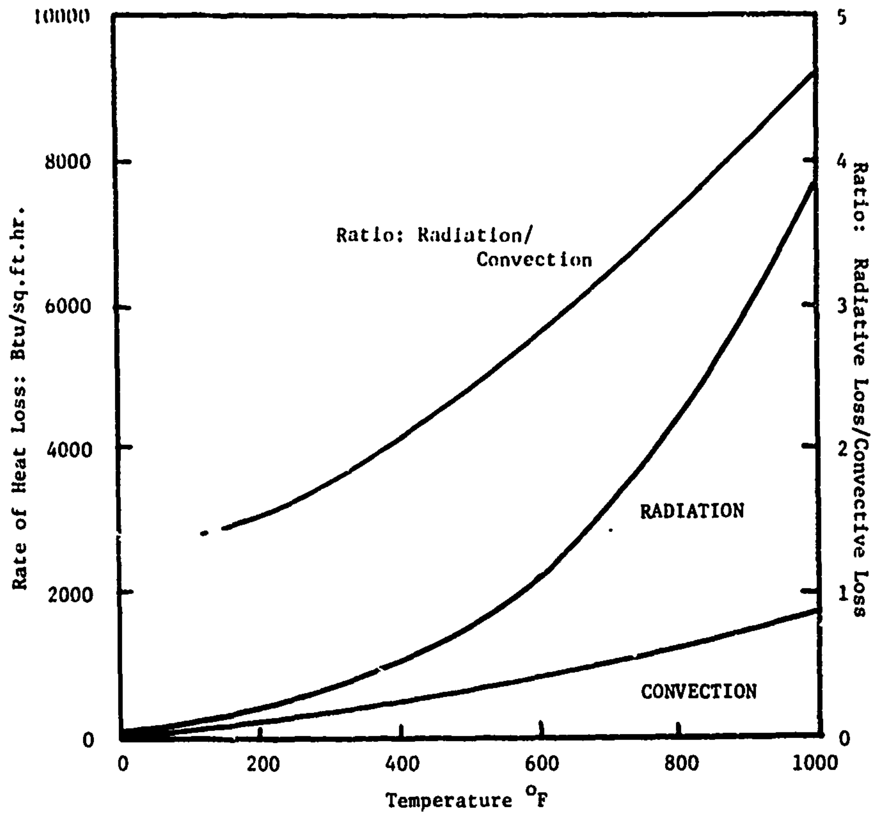

Figure III.1 Variation of Rate of Heat Loss from a Surface with Surface Temperature by Radiation and Convection: Including Ratio of Radiative to Convective Loss.

PART IV: POSSIBLE DEVICES

IV.1 Introduction - Stoves for heating enclosures have been in existence for so long now (decades running into centuries) that the possibility of inventing an entirely new natural-draft burner is remote. The best that can be hoped for in all probability is to find significant ways of improving on existing devices.

In considering possible improvements, these will be discussed under two general heads:

(1) Heater modifications and improvements to obtain more uniform and efficient combustion.

(2) Fuel modifications that might enhance handleability, combustibility, and/or reactivity of the fuel.

IV.2 Basic Requirements -

(1) All heating stoves must conform to certain unalterable requirements. The stove must form an enclosure in which the fuel is burned, with apertures for: (i) supply of fuel; (ii) supply of air (if not premixed with fuel); (iii) removal of hot exhaust gases; and (iv) removal of other wastes, if any, (ash, etc:). The enclosure can be a single chamber or multiple chambers. The walls or shell of the enclosure must be built of refractory brick or metal to withstand the heat. The fuel and air can be supplied separately or together from the top, the sides, or the bottom, through single or multiple pipes and apertures. In the provision for fuel feed this can be either: (1) batch (as in coal or wood firing); or (2) continuous or semicontinuous (as in liquid fuel firing or hopper units). An appropriate fraction of the heat in the flame and exhaust gases must be transferred through the enclosure shell and/or through the stovepipe to the surroundings being heated. For that last purpose, the lower the enclosure shell temperature the larger the surface area required to match some given level of heat output.

The choice of batch or continuous feed is largely limited by choice of fuel and there are then subdivisions of each of these alternatives. In batch feeding of solid fuel, this can be fed either onto a grate (allowing the use of underfire air) or the fuel can lie on a solid hearth. In continuous feeding of high volatile liquid fuels there is the choice of vaporizing in the combustion chamber, or before entry to the combustion chamber.

(2) The number of degrees of freedom that would permit permutations of the above basic requirements is then very limited; the permitted permutations are really confined to: (1) number and shape of chambers; (2) number and location of air supply ports (and to a lesser extent the size of

the ports); (3) the number and location of the fuel supply ports in the cases of vaporization prior to the combustion chamber. In the cases of solid fuel firing or of liquid fuel firing with evaporation in the combustion chamber, there is really no choice at all for the fuel location: it is compelled by gravity to flow or travel to the bottom of the chamber. Top hoppers with gravity feed (Sec. III.8(c)) are fairly common for burning coke, but these are not suitable for tents because of the specialized fuel. What may never have been considered before is flow of liquid fuel down a vertical or inclined plate, though what advantages this might have (if any) can only be discovered by experiment.

The fuel location likewise imposes constraints on the air supply locations. Sufficient air has to be mixed with the fuel vapors and directed through, or over, any char residue formed during combustion. Consequently, when the conventional designs with fuel at the bottom of the enclosure are used, most of the air ports will be located preferentially in the lower half of the enclosure. In the particular case of solid fuel firing on a grate, the limit condition is for all the air to be supplied as underfire air.

With grate firing it is common to use a two part grate with one port movable from the outside to riddle the ashes. In that case an ash pit must be included under the grate. However, if most or all the air is best supplied <u>over</u> the solid bed, relying on diffusion or directed jets to carry air into the bed, the grate is no longer necessary (except for providing storage space for ash under the bed) and the fuel can lie equally well on ashes. These ashes then only have to be periodically raked out by a flat hook or claw. Noncharring fuels, of course, require no ash removal ports.

(3) The location of the exhaust port is conventionally, but not invariably, near the top of the stove. However, when it is near the bottom of the stove, this generally means that there is one or more dividing partitions. If reaction continues up to the point of gas exit, the passages formed can be regarded either as an extension of the combustion chamber, or as separate (multiple) chambers, according to personal preference. If reaction stops some distance before the point of exit, the passages are called flues. The exact designations become a matter of definition, but this does not alter the functions of multiple chambers and internal walls, whatever those supposed functions may be. Apart from a few obvious cases where partitions clearly direct the air flow to where it is needed, and prevent short-circuiting, and the particular case of the Aladdin burner, there seems otherwise to be three intended functions. (i) The hot gases can be brought more directly into contact with a larger fraction of the containing shell thus (hopefully) promoting heat transfer to the outside. Narrower passages increase the velocities, but since the hydraulic diameters drop at the same time the net effect on Reynolds number is usually for it to drop so that the heat transfer coefficient may also drop, thus defeating the purpose of the higher velocities. (Since the flow is likely to be in

the transition region in the narrower passages, and the passages are odd shapes, the effect on the heat transfer coefficient is difficult to forecast.) (ii) Different passages or chambers usually cause a change of direction of flow and this is believed to improve mixing. However, if mixing needs to be improved at the end of a chamber, it needs it just as badly at the beginning of the chamber, and this is better achieved by improved injector design for the secondary air. Furthermore, the low Reynolds numbers in the narrower passages do not lead to any expectation of reasonably improved mixing. This is often better achieved by passing the gases through a venturi section. (iii) Finally, there can sometimes be some merit in carrying the hot gases down behind a dividing wall so that heat transfer through the wall will help to heat the fuel on the other side. (As a fourth possible reason there is also a suspicion that some designers believe that dividing the combustion volume into passages increases the retention time for the gases, thereby giving them more time for reaction, but this is obvious nonsense unless the partitions serve to eliminate substantial stagnant regions.)

(4) In summary, therefore, except for the Aladdin burner, partitions creating multiple chambers may have a little value (or be essential for top gravity-feed hopper stoves), but the value is mostly negated by the extra weight and complexity of the unit and the likely need to use high temperature refractory metals to withstand the heat, which are likely to be expensive. If we restrict attention to single chamber units, the possible permutations on design are therefore restricted to (1) the method of air supply; and (2) the method of fuel supply. Nevertheless designs based on multiple chambers might have value in special cases (see Sec. IV.4) and the idea should not be discarded at this stage. However, the priority of attention given to multiple chamber units should be quite low except where logic clearly leads to such design, as for example indicated in Sec. V.4.

IV.3 Single Chamber Units with Conventional Feed - The single chamber unit with conventional feed is one in which the solid or liquid fuel lies on a grate or a hearth or in a trough at the bottom of the combustion chamber. The air in existing designs is supplied in multitudinous different ways; but the best evidence is that no known devices create a "backmix" flow in the space over the bed as discussed in Sec. III.4. There is no doubt that existing designs work, but they are often somewhat tempermental, somewhat inflexible, and given to producing smoke when badly operated or overfired.

If improvements on such conventional devices are possible, the only degree of freedom (with conventional feed) is to permute the method of air supply. There is no firm evidence that this will necessarily improve matters, but the best evidence is that it will probably do so. Changes in overfire air pattern are therefore proposed as a prime factor for investigation. Three arrangements in particular are suggested for examination (Sec. III.6): a single vortex, a double vortex, and backflow jets. Other arrangements may suggest themselves as experiments proceed. Questions that arise are: (1) Will such arrangements improve performance (flexibility, change in burning rates, etc.)? (2) If they do, is natural draft sufficient to drive the necessary mixing? (3) Can the more complex design be incorporated into a rugged, mobile unit, or will it increase the weight and maintenance requirements beyond acceptable levels?

If improvements are obtained, e.g. wider limits of smoke-free operation, it will substantiate the expectation that overbed mixing is at present inadequate. Particularly if smoke is suppressed it will imply improved burn up and/or prevention of liquid particulates (which are the primary constituents of most smokes) being converted into solid particulates.

IV.4 Aladdin Multiple Chamber - This unit is discussed as a special case because: (1) it utilizes multiple chambers with relatively flimsy partitions, (2) it permits clean-burning smoke-free (blue flame) combustion over a range of firing rates, (3) the evidence is that the multiple chambers are necessary for the smokeless, blue-flame combustion, and (4) the unit is produced as a portable stove for military use.

The significance of this unit is that it represents the how of a practical means of achieving odorless blue-flame combustion in place of the usual yellow-flame combustion which is often odorous and has considerable smoke forming potential. The particular reasons why the particular chamber and air configuration achieves the stated result is at present unknown. If we did know the reasons, such information would very probably be invaluable in improving design of other units. It would be particularly advantageous to know: either whether it corresponds to the stirred reactor configuration discussed above; or whether it achieves the end result by different means. In particular, the present impression is that the stirred reactor configuration prevents smoke and low combustion efficiency by burning up already-formed smoke "precursors"; but the Aladdin configuration may prevent the smoke precursors in the first place.

Comparing the Aladdin burner with other units we note: (1) that the Aladdin burner is designed for volatile fuel oils (kerosine), and it is known to be superior to conventional units for this fuel alone; (2) conventional stoves will mostly handle other (higher carbon) fuels, which the Aladdin so far as is known will not; (3) the nonluminous combustion in the Aladdin burner reduces the radiative thermal loading on the chamber partitions: if such a unit is fired too hard or with a higher carbon fuel and at higher combustion intensity, it is almost certain that the relatively flimsy partitions will not survive the thermal stresses. However, this conjecture is again a matter requiring test by experiment.

In summary: the Aladdin burner is one that, in its present form, is probably too flimsy (easily rectified) and too limited on fuels for choice as a replacement bunker heater, in spite of its evident advantages. However, it generates the following critical questions: (1) Can the principle (whatever it may be) be adapted to other fuels?; and (2) Can investigations establish what the principle is, so that adaptation to other, more rugged, devices becomes possible?

IV.5 Unconventional Fuel Feeds - The discussion in the preceding sections presupposes the conventional arrangement whereby fuel flows or falls rapidly to the bottom of the combustion chamber. However, unconventional arrangements can be imagined.

IV.5.1 Liquid Fuels - For free-flowing liquid fuels, these could be allowed to flow down a vertical or an inclined plate. The plate could be the wall or walls of the combustion chamber, so they would be kept cool, which could be advantageous in some circumstances. The value of such a feed could be that the area of plate covered by a liquid film would increase with feed rate so that evaporation would always match exactly the feed rate, until the film finally reached the bottom of the plate. If the plates formed a cone or a wedge, then surplus would fill up the bottom and again it could provide a variable surface area for evaporation control.

As another variant, the inclined plates could be vertically ridged, with the fuel flowing down the troughs and air supplied through holes along the ridges. This would improve the vapor/air mixing.

The above proposition was generated solely as the result of asking: How else can fuel oil be supplied? Until it is tried it is not possible to say whether such an arrangement would have any advantages from the combustion point of view. There may be problems from carbon forming on the plate. It does, however, have possible advantages over the pool or pot-type stove from the point of view of safety since the film of liquid running down the plate does not represent as big a fire hazard, if the unit is overturned, as does a pool.

IV.5.2 Viscous and Solid Fuels - The inclined ridged plate might also serve double duty for burning very viscous oils, if some means could be found to feed such fuels to the top of the inclined plate in the first place. If the viscous fuels could be formed into "gelled" or even frozen sticks then they could be fed in by hand, as required, just as if they were solid.

Solid fuels such as wood or coal might also burn in the same unit although arrangements would have to be made for ash removal. However, with the plates forming a cone or a wedge, it is almost certain that some of the fuel would bridge across and prevent complete utilization of space. This could be avoided if the plate could be removed and replaced flat on the bottom of the unit. It would then serve either as a grate, or as a series of troughs for free-flowing liquid fuels.

Such potential versatility does therefore recommend the perforated ridged plate for investigation as a possible alternative to the conventional cast-iron open grate.

IV.5.3 Other Liquid Feed Devices - Liquid fuels could also be fed through porous cups of either metal or refractory at the tip of the feed pipe. Such tips might then act as wicks. Kerosines would flow fairly freely. Other fuels would need higher temperatures. If the fuel supply pipe formed a coil inside the combustion chamber, it would be heated by the flame it enclosed. To allow more viscous fuels to be given higher levels of heating, an internal sliding chimney could be raised or lowered to provide the desired degree of heating. A disadvantage could be that if the tip overheated, formation of carbon on the sintered cup would close the pores and block the fuel flow. However, with fuels stable enough to vaporize without cracking and blocking the feed tube, there is not necessarily any need for a diffusing tip. An open jet would be just as satisfactory, with vapor spraying out. Such devices do exist already on the commercial market, but they are liable to erratic flow.

Such means of vaporizing the fuel might therefore be considered, but expected problems of fuel breakdown with clogging, starting difficulties when cold, etc., would not indicate that any investigations on these lines should be given high priority.

IV.6 Fuel Modifications - The discussions to this point has been predicated mostly on the assumption that fuel supply to the combustion chamber will always be accomplished by one means or another. However, some problems have been indicated. At normal temperatures fuels can be classified as:

(1) Free-flowing fluids

(2) Viscous fluids

(3) Solids

This ordering also represents decreasing H/C ratio and volatility.

Supply of free-flowing fluids is no particular problem; they are also the more volatile and more easily burned. Solids are also fed relatively easily, although they present virtually new and unsolved problems in combustion if they melt. The viscous oils represent a new problem in feeding (their handling in commercial practice is discussed in Sec. III.8(d)).

In cold to very cold (sub-zero) weather, most of the free-flowing fuels become too viscous to flow readily. The fuel cans then have to be kept inside the tents and huts to keep the fuel sufficiently warm to flow, and this creates a fire hazard. However, it presents a particular problem just after setting up camp and there is no warm fuel available for initially firing up the heater. An alternative solution might be to take advantage of this behavior, and arrange to feed the fuel in solid sticks after suitable modification.

The simplest modification that can be imagined is to pour the fuel into plastic or paper or waxed paper bags of suitable size when it is still reasonably free flowing and freeze the sticks. This method might be open to the following objections. The fuel sticks may freeze to each other when stacked, so they would not easily come free from a pile for use. They would also have to be kept stored outside till actually used. In use, as the bags burned off and the sticks melted, the heater would be flooded with fuel. This means that melting might have to be carried out in a side reservoir, with obvious complications from the paper or plastic containers.

These problems would be obviated if the fuel stick could be stabilized so it would not melt on heating (at least at normal temperatures). Treatment to turn the fuel into gel, or even to attain some degree of polymerization, should be satisfactory if this can be achieved. If it then burns as would a piece of coal or wood then fairly conventional stove designs could be used.

A critical uncertainty in designing heaters to operate on the more viscous fuels (including wastes such as crankcase oil) is the lack of information on its behavior during combustion in natural draft heaters. Waste oils are frequently incinerated but usually with supplementary fuel, or else in pits where any char not burned up is either left there or covered over. Some information is now needed (it may be available but has not been located) on the behavior of such materials in conventional (or unconventional) stoves. Questions to be answered include: What fraction is pyrolysed or evaporated off? How porous and friable is the coke residue? Does it go through a heavy tar stage that is inclined to block all air holes? Answers to these questions will be necessary to know what degree of modification may be required in stabilizing the liquid fuels as solid sticks.

The preferred behavior of the sticks in combustion would be for them to pyrolyze or evaporate off fairly light volatile components, with the residue forming a porous coherent char which would then be of quite high reactivity. There is still a problem of continuous feed to replace hand feeding if possible. If adequately stabilized, the sticks could be stored inside as well as outside, and would represent a much reduced fire hazard. Waxed wrappings might also act as a touch paper for starting the fire with a match even in extreme cold.

Required stick sizes can be estimated as follows. If the calorific value of a(waste) fuel is taken as 15,000 Btu/lb., a small tent would require about 3/4 lb. per hour. A fuel stick of one inch diameter and one foot length (or two inch diameter and three inches long) would weigh about five ounces. The fuel rate therefore would be just over two sticks per hour. A

24 hour supply would be 48 sticks which would occupy a box of dimensions roughly 7" x 7" x 12" (approximately 1/4 cu. ft.). These figures would be multiplied by 10 for a larger hut. A tent stove of 1/2 cu. ft. of combustion space could therefore carry one to two days supply of fuel inside it during transportation. Further sticks could be carried in the stove pipe.

IV.7 Stove Shapes - Shape is one additional degree of freedom that could have some influence on combustion efficiency; it will also have some effect on the thermal efficiency because of the increase in surface to volume ratio for a given volume as the shape departs more and more from an equivalent sphere. From this point of view, either a tall narrow cylinder, or a unit with a narrow rectangular cross-section would be advantageous. Both would be less stable, (more liable to be knocked over) and therefore a greater fire hazard, than a more squat unit; but both would have the advantage that air jets for mixing would have a shorter distance to travel, thus requiring less draft for a given proportional penetration. However, if single or double vortex flow is effective in reducing smoke emission, this is more readily accomplished in the cylindrical unit than in the narrow, slab-shaped unit. The slab-shaped unit, on the other hand, might be a more appropriate shape to investigate the effect of back flow jets, and it would better accommodate a sloping plate for alternative fuel supply (an opposing argument to the high surface volume ratio is developed in Sec. V.2 Comment 3. It depends on being able to raise the stove shell temperature).

IV.8 Radiation Panels - To increase the thermal load on the heater and thus improve its thermal efficiency, the incorporation of radiation panels was indicated in Sec. III.11. Such panels would also increase the radiation flux, as indicated to be desirable in Sec. I.4.

To have any worthwhile effect, the radiation panels would have to be at temperatures lying at least in the range 700 to 1,000°F, and preferably higher if possible. However, such specification also means that the flame gases heating the panels must substantially exceed those temperatures (preferably by at least 500 to 1,000°F). Since average temperatures are estimated to be only about 1,500°F, and gas exit temperatures 200 or 300°F lower, then the radiation panels must be located in the vicinity of the peak temperatures. With good vortex mixing, peak temperatures in the range 1,500 to 2,000°F should be easily attainable. These are higher than the experimental smoke flame temperatures mentioned in Sec. III.9.2(i), but the predicted possible temperatures at higher fuel rates at 50 to 100% excess air were 2,400 to 3,000°F; so with the correct fuel rate and mixing effectiveness, the proposed peak temperatures are by no means impossible.

Location of the radiation panels near the peak temperature region introduces further constraints on shape and dimensions of the stove. To make sure the panel is well heated, it might be desirable to engineer impact of the flame on the back of the panel by introducing suitable internal partitions: this would increase the convective transfer. If the back of the panel is always red hot no problems of fuel cracking and carbon buildup need be expected. The easiest method of achieving flame impact would be to make the radiant panel the top of the stove (and maybe turning one corner to travel part way down one side). A radiant top of this sort would then do double duty as a cooking surface. The objection could be that the radiant flux would then be directed mainly at the roof; however, lightweight reflection plates might compensate for this.

A design to produce flame impact on the top of the stove would mean something like a two chamber design, with the two chambers formed by a vertical partition. The flame path would be up one side of the partition, over the top (scrubbing the underside of the radiant panel at the same time) and down the other side. This would form an inverted U-shape. The chamber shape would then, in effect, correspond to a thin slab or a tall cylinder (Sec. IV.7) but bent in the middle to form the inverted U. There is here a clear logical reason for a double chamber unit.

The use of a radiant panel and possibly an internal partition reintroduces the materials problem. This really needs to be considered in detail quite separately. Briefly, however, refractory metals capable of withstanding the temperatures indicated do exist. This is by no means an inseparable problem. The necessary metals may be expensive, but the trade off against reduced fuel consumption may make it worthwhile. This is a factor for future determination.

IV.9 <u>Heat Exchangers</u> - Overall thermal efficiency may still depend ultimately on some degree of heat recovery from the stack gases. If this may be accomplished easily it deserves quite high priority. The very simplest heat exchanger would be to have the stovepipe surrounded by an air supply pipe, with the hot gases going up the center and the cold air coming down the annulus between the pipes. Or, a single pipe could have a vertical partition, forming two cross-sectional semicircles.

A rough estimate of the required heat transfer area follows. The mass flows of air and combustion products, and their specific heats, are very close, and for an approximate calculation can be considered equal. This simplifies the heat exchanger equations considerably. For a heat exchange area A (sq. ft.), mass flow M (lb/hr), specific heat C (Btu/lb.°F), the standard counter flow heat equation may be written

$$Ah/MC = (T_{1i} - T_{1o})/(T_{1o} - T_{2i}) \qquad \text{(IV.1)}$$

Where T is temperature, and subscripts 1 and 2 refer to the two fluids with i and o referring to inlet and outlet temperatures respectively. Fluid 1 heats fluid 2. If the gas inlet temperature is 1,400°F and it is cooled to 700°F, $(T_{1i} - T_{1o}) = 700$. If the cold air inlet temperature is 0°F, then $(T_{1o} - T_{2i}) = 700$. Therefore $(Ah/MC) = 1$, with this ratio rising as the cold air inlet temperature rises, and falling as T_{21} falls. For the smaller (tent) heater, the estimated air requirement is 150 cfh, or 12/lb. The specific heat is about 0.25 so MC is 3. Since h is about 1 Btu/sq.ft.hr. ^{o}F, we find A = 3 sq. ft. If the heat exchange surface is a circular pipe of diameter d (ft.) and height 10 ft., its area is ($10\pi d$) = 3. Evaluation yields a diameter of a little over one inch. Consequently, a stovepipe of 2 to 2.5" (Sec. II.6.3) would provide more than adequate heat transfer surface; alternatively it provides some margin if the value of h is not as great as expected.

The requirements for the larger unit are also compatible. The mass flow is ten times greater (in a larger diameter pipe, so h may double). The area required is therefore closer to 30 sq. ft. If h doubles, the required pipe diameter would be nearly 6", which is a little larger than originally proposed (Sec. II.6.3); but the larger unit could well justify a taller stack (say 15 ft.). However, there are sufficient uncertainties in the calculation that the predictions should be regarded as having order of magnitude accuracy only at present. The calculations do then show that concentric tubes would probably provide roughly adequate heat exchange surface. A disadvantage could be the increased diameter required which makes the unit that much more bulky and therefore less portable. Again this may be a matter of trade off between convenience and efficiency, with the balance lying to one side or the other, depending on current circumstances.

A minor point should be noted concerning light-up. When both tubes are cold, both can act equally well as chimneys, and the exhaust gases would tend to use both. This tendency can be eliminated by a short-circuit opening at the bottom of the air supply pipe to the tent or hut enclosure, opened temporarily. Once the stovepipe has heated up it should "draw" properly, and the combination tube would act in fact as the classical U-tube representation for calculating stack draft (Sec. II.7.2) is supposed to act. Another minor point to bear in mind is that the opening for the air inlet should be sufficiently displaced from the gas exit that the exhaust gases are not drawn into the air inlet.

IV.10 Forced and Induced Draft - All the preceding discussions have been based on the assumption that the only driving force available for the air flow is natural draft, due to gravity (buoyancy effects). It is true that this is the only force that can always be depended on; it is also true that no mechanical devices can be assumed to be available. However, both forced and induced draft can be obtained from wind flow. If wind flows through a venturi at the top of the stack always oriented to face upwind, this will provide some induced draft. If the air ports for the air supply stack also face upward, the wind velocity by stagnation or partial stagnation will increase the upstream pressure and provide some forced draft. The increased or F. D. pressure is effectively that which would be measured by a pitot-static. For velocity in ft./sec., the pressure in inches w. c. is given by

$$h = 2.4 \times 10^{-4} v^2 \qquad \text{(IV.2)}$$

A pressure of 0.1" w. c. is developed at about 15 mph, so below this velocity the wind effect is secondary or negligible. Above that, however, it rises quickly (with the square of the velocity), reaching 1" w. c. at about 45 mph. At velocities between 20 and 30 mph, it would probably prove necessary to choke the air flow. This could become too great altogether, leading to low efficiency due to high excess air just when high efficiency is most needed (because of extra cooling). There is probably not too much point, therefore, in trying to harness this effect beyond a modicum because it could cause more problems than it solves.

IV.11 Fuel Storage - As a minor item concerns a possibly convenient mode of fuel storage for any solid fuel sticks (Sec. IV.6) if these are fairly regular in size. A small diameter tube attached to the stovepipe (but not heated thereby) could carry ten such sticks, so they would be easily accessible by sliding down the inside (if hang up due to freezing is not a problem). They would be out of the way, and present a reduced fire hazard. Again, however, this might present more of a complication than convenience would justify.

IV.12 Summary -

(1) The balance of argument supports the expectation of being able to improve existing stove and heater designs, but probably not to be able to develop a radically new design.

(2) Analysis is in favor of a one chamber, or at most a two chamber unit, with dimensions narrow and long rather than tending to cubical or spherical unless the whole unit can be compacted, with higher combustion intensities and stove shell temperatures. This is to reduce the mixing path length for secondary air, thus requiring less draft for mixing, and to increase the surface to volume ratio for increased heat transfer.

(3) If fuels are to be solid or free flowing liquids, a perforated ridged plate might possibly be able to accommodate both. If the plate can be inclined, this might reduce the fire hazard from the free flowing fuels. Experiments on such units are suggested.

(4) Fuel modifications to enable viscous liquid fuels to be transformed to fuel sticks that would at worst melt only in the stove, and at best would pyrolyse to form free burning volatiles and char, is also suggested.

(5) Provision of radiant plates seems feasible (if suitably cheap refractory metals are available). Overall, the expected heat transfer temperatures should be in the region of 1,500°F. Gas exit temperatures would be lower, by 200 to 300°F; and peak temperatures would be higher, by at least an equivalent amount.

(6) A heat-exchanger stack formed of two concentric pipes with the exhaust inside and fresh air outside, looks feasible and could quite probably increase the thermal efficiency considerably.

(7) Utilization of the wind for forced draft, however, is probably not worth pursuing.

PART V: GENERAL SUMMARY AND RECOMMENDATIONS

V.1 Objectives - The principal objective of this study was to determine and characterize sources of efficiency and inefficiency whilst heating tents and bunkers under semiarctic and arctic conditions by small, natural-draft, portable stoves; coupled with recommendations on possible improvements, and procedures to test and implement the possible improvements. Efficiency can be characterized under four heads:

(1) Combustion efficiency, relating to completeness of combustion and total burn-up of all fuel supplied.

(2) Thermal efficiency, determined by the lowest stack gas temperature that will maintain the minimum draft requirements.

(3) Use efficiency, controlled by such factors as the radiation/convection balance; tent or bunker insulation; air infiltration characteristics; thermal gradients in the bunker; and the total Heat Demand of the enclosure.

(4) Design efficiency, concerned with optimizing grate and overfire air designs; shape and internal configurations; heat exchangers; fuel feeders; radiant panel locations; stack design; etc:.

Comment: There is obviously some degree of overlap between categories. Item (4) in particular overlaps with all three others.

V.2 Data Summary - Table V.1 is a summary of all numerical estimates appearing in this Report. The majority of the values are believed to be probable average values of existing units, or best values of existing units if carefully operated.

Comment 1: The stack exhaust temperatures do seem to be high, but they would, of course, be substantially reduced as the excess air is increased (see Fig. II.1). The problem is that the temperatures never seem to have been measured, nor can there have been much incentive to do so. Furthermore, measurement by bare thermocouple, the likely favored method, is quite inaccurate. H.V.T.'s must be used instead (suction pyrometers).

Comment 2: The radiative fraction of heat loss from the stove shell (75%) also appears high in view of the comments of investigators calling for more radiation (V. Appendix I). The source of the high estimated value is probably the possibly optimistic assumption of stove size required. A 10% increase in a linear dimension of the stove will increase the surface area by 30% and drop the surface flux density from 2,000 Btu/sq. ft. hr. to 1,550 Btu/sq. ft. hr. This would drop the required stove surface temperature to 400°F, and the radiation/convection balance to 2 to 1, down from 3 to 1. These changes would make little difference to the internal heat transfer estimates.

Comment 3: The estimates in Comment 2 would indicate the value of stove designs that minimize surface to volume ratio, so long as the stove shell temperature can be raised, in apparent contradiction to the conclusions of Sec. IV.7 Minimum surface to volume ratio would require compact designs (with higher combustion intensities) to keep the secondary air mixing path lengths short, as required by the argument of Sec. IV.7.

V.3 Unacceptable Fuels and Firing Methods - The following fuels and firing methods were considered unacceptable for serious consideration for reasons appended. The judgements are not absolute; exceptions are considered in the Comments following.

(1) Fuels

(i) Gas - generally unavailable in anticipated operational areas. Bottled gas could represent an unacceptable weight penalty. In steel cans the effective transportable weight of liquid propane is approximately 10,000 Btu/lb. In aluminum cans it is about 15,000 Btu/lb., which is not an excessive penalty. This lies between coal and oil.

(ii) Coke - also generally unavailable in anticipated operational areas. Again, other fuels have a higher Btu/lb.

(2) Firing Methods

(i) Wick - requires a clean high volatile fuel: units will not generally operate on emergency substitutes.

(ii) Prevaporization - as for (i) above: this conclusion is subject to revision if other fuels can be modified to flow freely and be heated without cracking.

(iii) Droplet Sprays - (domestic type burner): suitable for a wider range of liquid fuels, but utilizes air or pressure atomization which requires mechanical or electrical pumps.

(iv) Hopper stove (for solid fuels) - most designs can only use coke. Thring stove can use coal. However: internal construction adds unacceptable weight penalty unless weight can be reduced by using lighter weight, thinner refractory metals.

Comment 1: With forethought in design, all stoves could be arranged to accommodate prevaporizing (premixed-type) devices, or wick units, to be installed in high-priority or emergency situations with the necessary high volatile fuel made available by special supply.

Comment 2: In the exceptionally rare circumstances that pipeline (or bottled) gas happened to be available, the basic stove design should again be such that temporary installation of gas burners is quick and simple.

Comment 3: If the basic stove design can accommodate wood, coal, and some solid wastes, it should be capable of burning coke with maybe some increased draft or alteration of the overfire to underfire air ratio. Such capacity should be investigated experimentally as part of any proving tests.

Comment 4: Although the target of this appraisal is in connection with heating under subarctic and arctic conditions, any improvements should have general applicability. A worldwide survey of fuel availability and pattern of fuel use should be made so that advantage could be taken in any special circumstances of this knowledge from the point of view of provision of special adapters or inserts.

V.4 Suggested Design and Firing Characteristics - The characteristics of any new design must approximately match those listed in Table V.1 if the new unit is only to perform as well as existing designs. Design improvements must then be able to increase or decrease (as appropriate) those values listed in the table as inequalities or as extrema. Specifically, the objectives should be to operate with: minimum excess air; minimum stack gas temperature (consistent with draft requirements); maximum stove shell temperatures, thus maximizing the radiation/convection loss ratio. This last is likely to mean quite a high gas temperature entering the stovepipe, thus requiring a stack heat exchanger to obtain good thermal efficiency and minimize the stack gas exit temperature. Finally, the combustion volume should be as small as possible, with smallest surface to volume ratio consistent with overfire air penetration requirements. When the air penetration requirements are on the margin of no longer being satisfied, this marginal condition determines the upperbound limit to the smallest (cross-channel) dimension. In any further increase of total volume, this upper limit to the smallest dimension should not be exceeded. With adequate heat recovery on the stack, doubling the combustion channel back on itself to form an inverted U-shape will help to preserve the minimum surface to volume ratio, and hence the maximum stove shell temperature, without sacrificing overall efficiency. If stack heat exchangers are not used, the stove shape should be designed to increase the surface to volume ratio (and thus the heat transfer area) to achieve good thermal efficiency, even though this may be at the expense of stove shell temperature, and hence the radiative heat transfer component.

The above summary itemizes the following components: The number of chambers should be one, or two at most; with shape determined jointly by air mixing and heat transfer requirements, including the presence or absence of special radiant panels in the shell of the stove and/or a heat exchanger in the stack.

Other items considered were: (i) the desirable inclusion of a separate outside air supply pipe as an integral component of design, with a cross-link between the stack pipe and air supply pipe that would contain both an atmospheric and a hand-operated damper; (ii) the desirability of developing a multifuel grate, such as a ridged grate with air supply holes on the ridges, to be used either horizontally or inclined, with either solid or liquid fuels; and a possible means of increasing the fuel scope still further, by examining the feasibility of developing modified fuels by physical or chemical means that would in effect stabilize liquid fuels as solid fuel sticks.

Comment 1: The above considerations still are based on the assumption that free-flowing liquids can be continuously fed, with fairly precise variation of feed rate; but that solid fuels (including the conjectural fuel sticks of stabilized liquid fuels) can only be fed in batches.

Comment 2: The positive fuel feed control mentioned as desirable in Paragraph 3 of the Task Description (Task No. 05-S-71: 1 Mar 71) is still therefore an unsatisfactorily resolved item.

Comment 3: If solid sticks of stabilized liquid fuels can be developed that will mainly melt on heating, it might be possible to control the fuel supply rate by stacking the sticks one above the other, with gravity assisting variable weights to push the sticks down into a cup inside the stove, with variable (preheated) air jets directed into the cup to control the rate of melting.

Comment 4: The items otherwise listed in Paragraph 3 of the Task Description that have been considered include: multifuel capability; and optimum heat transfer. Liquid fuel preheat to improve combustion efficiency is not recommended because of the risk of vaporization with erratic feed of the more volatile fuels, and the risk of cracking, with carbon formation that would block fuel lines in the case of the less volatile (higher carbon) fuels. Heat recovery with air preheat is a better method.

V.5 Comfort and Heat Demand - Further comment on this topic is provided in Appendix I. The salient points are that: (i) to reduce heat losses, tents and bunkers should be well insulated to reduce conduction losses and well sealed to reduce air infiltration losses; (ii) physiological (and combustion) requirements can be met by one to two changes of air per hour; (iii) high radiation levels from the stove or bunker heater are preferred, with the radiation flux directed horizontally rather than vertically; (iv) to reduce radiation losses, the inside of the tent enclosure should be light in color and preferably with reflective coatings; (v) the chief defect, leading to discomfort, was found to be the steep vertical temperature gradient, with low air temperatures, at foot level where the surface to volume ratio of the body is high.

Comment: A realistic viable air temperature for a subarctic or arctic tent is still undetermined.

V.6 Safety -

V.6.1 With respect to Fire Safety, the hazards are: (i) stoves *in situ* with flammable material falling on them; (ii) stoves being overturned and falling on or spilling out burning fuel onto flammable material; (iii) liquid fuel feed lines disconnecting or rupturing.

Comment 1: Designs favoring small, compact, low and squat heaters reduce chances of stoves being overturned. Designs favoring hot stove shells or radiant panels increase chances of material falling on the stove being ignited. Designs would therefore better favor cooler shells with radiant panels protected by a coarse wire screen, possibly including closable reflector doors.

Comment 2: The disconnecting or rupturing of liquid fuel feed lines is probably an unpreventable hazard. The hazard does not exist with solid fuels.

V.6.2 With respect to Health Safety, the prime hazard is carbon monoxide.

Comment: The more fully sealed a unit, with separate access pipes to the outside for supply of combustion air and ejection of combustion products, the less chance of gases escaping from the stove to the living space. The better the internal design, including the mixing requirements, the less chance of there being dangerous concentrations of carbon monoxide in the stack gas.

RECOMMENDATIONS

1. OBJECTIVES - The purpose of the following recommendations is to provide an R&D and Task sequence for improving natural-draft bunker heaters in all possible pertinent aspects.

Comment: It should be emphasized that the natural-draft combustor is the most difficult of combustion devices to design and operate since the only power source is gravity.

2. PRIORITIES - All recommended Tasks have been assigned to one of four groups. The groups are arranged in decending order of priority (I high to IV low). Within each group the order is a suggested but not strongly recommended order of priority. The basis of choice is ease of developing the information, and immediate pertinence or applicability of the information developed.

Tasks Group I - Development of Data Norms

Objectives: To establish an accurate data background for reference and comparison.

I.1 Worldwide Fuel Patterns - A map should be prepared showing local relative availability of possible stove fuels. The map should include wastes as possible sources (including crank case oil, refuse, etc.).

I.2 Fuel Priority Assessments - The priority of fuel over other material being transported should be assessed as a function of local availability, temperature, and weather conditions. In the tropics, for example, fuel could rate a very low priority (needed only for cooking); but in extreme arctic conditions when sheer survival becomes, even temporarily, the only objective, fuel could take top priority over, for example, ammunition and even food.

I.3 Weighted Fuel Pattern - The map of local relative availability of possible stove fuels should be weighted by means of the priority assessments to determine the most significant expected fuel sources and demands. This is part of the information necessary to be able to establish stove design priorities (liquid vs solid fuels, for example).

I.4 Viability Limits and Heat Demand - The desirable and permissible (optimum and minimum) viability levels - air temperatures, air change rates, radiation levels, etc: - should be established, and from these values the net Heat Demand for an enclosure, as functions of enclosure size, use, weather conditions, etc:. This information may already be available in existing reports; it may only require a literature search and collation of data collected.

I.5 Existing Heater Specifications - The actual performance of existing heaters should be determined experimentally with respect to all the data items listed in Table V.1. This will: (i) check and correct the estimates already made; (ii) provide an accurate data base for future comparison of improvements; and (iii) determine the extent to which the specified Heat Demand is actually met.

Tasks Group II - Existing Units Improvements

Objectives: To upgrade existing units by addition of one or more relatively simple improvements or modifications.

II.1 Investigate Overfire Air Pattern - Provide air ports in suitably chosen locations and retest unit performance - gas concentration, gas temperature, smoke point, firing range, etc:, for comparison with data norms developed under I.5 above.

II.2 Develop Draft Control Units - Add air supply pipe with atmospheric damper, cross-links to stovepipe, etc:, and investigate control of performance under reduced, fluctuating, and disturbed draft conditions. Compare with data norms.

II.3 Effect of Heat Exchange - Combine air supply pipe with stovepipe and investigate: (1) heat exchanger behavior; (2) preheat available; (3) draft modifications; (4) influence of air preheat on combustor performance. Compare with norms.

Tasks Group III - Bunker Heater and Components Redesign

Objectives: To develop information leading to redesign, with implementation of information by redesign where possible.

III.1 Combustion Chamber Shapes and Sizes - Different shapes, air port locations, sizes, etc:, should be investigated to determine an optimum design or limiting set of designs (possibly dependent on fuels). A limited number of possible shapes are described and discussed in general terms in this Report.

III.2 Grates - Different grates and internal fuel control devices (perforated ridged plate, horizontal or inclined, etc:) should be examined to develop improved units. Tests with different chamber shapes and air port locations may be necessary.

III.3 Fuels Handling - Improved methods of handling both liquid and solid fuels must be developed. In particular, a method for continuous solid fuel feed is required.

III.4 Materials Investigations - Information on the properties, mechanical, thermal, and corrosion resistance of different possible constructional materials for the stoves should be developed, with a view to use for the whole stove, or for radiant panels, or for internal divisions, etc:.

III.5 Fuels Modifications - The potential for modifying fuels to increase their fluidity in very cold weather, or form stable fuel sticks that will either melt on heating, or pyrolyze with char formation, etc:, should be investigated.

III.6 Redesigned Unit - A unit or units based on improvements arising out of all Tasks Groups II and III investigations should be constructed and tested.

Tasks Group IV - Background Research

Objectives: To provide a more fundamental basis for understanding the reasons for and theoretical limits to improvements developed by direct experiment under Tasks Groups II and III.

IV.1 Carbon Formation in Stoves - Permutation of Aladdin (blue flame) stove design to establish conditions under which luminous and nonluminous combustion, and smoke formation, do or do not occur; and from those and related investigations to develop as complete an explanation as possible to predict circumstances allowing or preventing carbon formation.

IV.2 Mixing and Air Penetration - Detailed investigation of mixing of jets and gas streams in pipes to enable prediction of optimum mixing pattern.

IV.3 Shape and Flow Pattern - Cold model investigations of existing and proposed stoves to determine their relative flow patterns. Comparison of mixing effectiveness ranking with combustion performance ranking. Correlation where possible of shape influence or desirable flow and mixing patterns. Use of such correlations to predict optimum shape.

Table V.1
Summary on Bunker Heaters

This Table summarizes all data developed in the te t of the accompanying report. The compilation represents what appear to be probable average values. of <u>existing units</u>, but including some predictions of potential improvements.

Function	Numerical Values	Report Section
Data Basis:		
Firing Density	20 Btu/hr.cu.ft.	
Excess Air	>50%	II.3[II.9.2(i)]
Thermal Efficiency	50%	I.2
Viable Enclosure Temperature	40°F	II.5(2)
Predictions: All Units		
Heater:		
Combustion Intensity	15,000-20,000 Btu/hr.cu.ft.	II/6.2/II.7.1
Max. Flame Temperature	1700-1800°F	III.9.2
Average Internal Temperature	1500-1600°F	III.9.2
Stack Exhaust Temperature	<1400-1500°F	III.9.2
Residence Time	3 to 4 sec.	III.4(ii)
Stack:		
Stack Height	10 ft.	II.7.2
Minimum Draft Required	0.075-0.1 w.c.	II.7.1
Minimum Permissible Stack Gas Temperature	700-800°F	II.8(9)
Heat Transfer:		
Stove internal (flame to shell)	{by Radiation ~50%	III.9.2(i)
	{by Convection ~50%	III.9.2(i)
Stove shell to enclosure	{by Radiation ≤75%	III.9.2(ii)
	{by Convection >25%	III.9.2(ii)
Enclosure (Tent or Hut)		
Heat Losses	{by Wall Conduction ~2/3	V Appendix I
	{by Air Infiltration ~1/3	V Appendix I
Rate of air changes for Comb.	once every 3 hours	II.3(3)
Rate of air changes for comb. and occupants	1 to 2 per hour	II.3(3) V Appendix I
Predicted Air Preheat Savings:		
Potential Recovery	50% of stack loss	III.10
Stack gas temp. reduction	to 700°F from 1400°F	III.10
Increased efficiency	by 1/3: from 50% to 67%	III.10
Increase in period for given fuel consumption	by 1/3	III.10

Table V.1: Data Summary (continued)

SIZED UNITS	Small	Large	
Capacity: Btu/hr	10,000	100,000	I.1
Air volume: c.f.h. (at 50% X's)	150	1500	II.3
Combustion Volumes: cu.ft.	0.5 (min.)	5(min.)	II.6.2
Dimensions [for cyl.ht.=dia.]dia.ft.	1 (min.)	2 (min.)	II.6.2
Velocities of gas flow: cu/sec.	7.5 to 10	15 to 20	III.4(ii)
Fuel Consumption: (lb/hr)			
Fuel oil at 18,000 Btu/lb	1/2	5	II.6.1
Coal at 12,000 Btu/lb	1	10	II.6.1
Solid waste at 6000 Btu/lb	2	20	II.6.1
Stove shell temperatures °F	500 (max)	700 (max)	III.9.2(ii)
Heat transfer flux density for for stove shell Btu/sq.ft.hr.	2000 (max)	4000 (max)	III.9.2(i)
Stack dia. (in.)	2 to 2.5	3 to 4	II.5.3

General Comment: Most existing bunker heaters tend to have dimensions minimizing their surface to volume ratios. Redesigns to be considered should include shapes either tending to increase the surface to volume ratio; or permitting higher stove shell temperatures [V.2. Comment 3].

Appendix I

Comfort and Heat Demand

This Appendix is supplementary comment to the material provided in Part I, Secs. I.3 and I.4, and in Part II, Sec. II.5(3), and is commentary on information given in three reports:

(1) "Thermal Performance and Habitability of Jamesway Tent" by John B. Pierce Foundation (1951) (Contract No. DA-44-109-qm-512).

(2) "Fuel Savings Resulting from the Use of Liners in Tents" by E. R. Gerhard and G. Cracraft (Aug. 1952) (Contract No. DA-44-109-qm-288).

(3) "Heat Retention Properties of Tent Liners" by C. J. Monego and H. J. Rasor (Nov. 1962) (Proj. Ref. No. 7-71-09-011).

These Reports became available after completion of Parts I and II of the subject study, and it substantiates with experimental evidence certain estimates and conjectures, as follows:

(1) The number of air changes per hour in the Jamesway Tent was found to be 2 1/2 per hour (c.f. Sec. II.5(3)).

(2) Combustion would require only one change every three hours. In the Pierce Foundation Report the combustion <u>and</u> occupant requirements are given as 1 to 1.5 per hour. The number of changes per hour recommended by Himus was 2 to 3 per hour. These figures are almost compatible.

(3) At 3 to 5 changes per hour the heat loss is estimated (Pierce Report) to match (approximately) the conduction losses through the tent walls. This is roughly comparable to the estimate that 2 or 3 changes per hour would correspond to 1/3 to 1/2 of the heat demand when outside temperatures are very low (Sec. II.5(3)). This does indicate that conduction losses are normally 2/3 or more of the heat demand. This is consistent with the conclusion in all three reports that tent linings can be appreciably effective in reducing heat losses.

(4) The reports also substantiate the value of radiation, together with reflecting surfaces (and light colors*) for the tent interiors. Radiant panels or sections on the stove or heater are also suggested, together with heated floors, which would be a significant problem. (Floor-directed radiation with reflectors might suffice.)

(5) The chief defects of the tents tested were: air infiltration, helping to create very cold floor temperatures and very marked temperature gradients; conductive and convective loss through the ceiling; and

absorption of radiation by walls and ceilings (partly rectified in some of the later experiments). Significantly, a low tent roof reduced the losses, presumably by reducing opportunities for setting up vertical buoyancy flows.

(6) The very low floor temperatures (35°F at 1" above floor, compared with 70°F at 4' and 85°F at 6') showed the heating defects, with the feet and leg areas which have high surface to mass ratio exposed to the lowest temperatures.

(Footnote)* A report in the journal "Research" about twenty years ago describes the effect of color on sensations of warmth and cold. Typists in a pool objected that their office, recently redecorated in pastel blue, was too cold, and temporarily the heating level was raised. On advice the office was redecorated in pastel pink; the typists then complained it was too hot until the heating was reduced to a level below what it had been before the first redecoration.

In commentary on these above points may be noted:

(1) The evident need for heating as close to the floor as possible. Therefore, tall and narrow heaters, or slab shaped heaters, can perhaps be designed with their longest axis horizontal instead of vertical; and/or: preheated combustion air could perhaps be arranged to flow through a horizontal heat exchanger on the floor, if the draft will permit.

(2) There could be some question about some of the conclusions in the reports, particularly regarding the "clo" factor requirements, since these seem to be based on air temperature requirements with radiation neglected. In that case, in the old T. B. Sanitoria above the snow line, lightly dressed patients exposed to sun and to very cold air (without drafts) would never have survived.

(3) The assumed viability temperatures of 65 to 70°F provide an interesting comparison with the Factory Act Regulations for heating of work enclosures in England. For sedentary (office) work the required range is 62 to 68°F. For light manual work the range is 57 to 62°F. For heavy manual work the range is 52 to 57°F. The novelist Alistair MacLean in "Night Without End" describes conditions in an arctic hut with temperatures ranging roughly from 30 to 40°F. The sources and reliability of his information are unknown.

(4) No estimates are made of the effects of occupant traffic in: (i) increasing the heat losses, and thereby increasing the average Heat Demand; (ii) mixing the air to produce more even air temperatures. At night, when traffic is marginal or zero, the occupants may be assumed to be in sleeping bags. However, the temperature results then emphasize the importance of using camp beds to take the occupants as far above the floor as may be practicable. This matter of the thermal gradient would seem to require fuller investigation.

(5) The third report cited, by Monego and Rasor contain substantial information of potential value on heat inputs and losses. Unfortunately, the data (on which estimates of savings are based) are quoted in units of "Btu/hr^{o}F". Since the temperature differences are unstated, the actual heat inputs and losses are unknown. The figures as given correspond in effect to total heat transfer coefficients but as a "lumped" parameter including all forms of heat loss. The actual Btu/hr. values would have been substantially more informative.