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RECOVERY OF WASTE HEAT FROM PROPELLANT FORCED-AIR DRY HOUSE

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This report describes the design, installation, and test evaluation of a heat pipe heat exchanger for a multi-base propellant forced-air dry house for the purpose of recovering heat from the hot air exhausted to the atmosphere during a typical drying operation. Detailed descriptions of the special safety and operating features of the recovery unit are also presented. Evaluation of the unit showed that approximately 62% of the energy used to dry multi-base propellants can be saved by the heat recovery technique employed. Energy/cost saving projections are based upon the installation of similar recovery units at a major Army propellant manufacturing

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facility such as Radford Army Ammunition Plant.

FOREWORD

In the early 1950's, this nation's demand for petroleum began to outpace its supply. Consequently, it began to import crude oil from foreign sources. This imbalance between internal supply and demand continued to increase until, by 1973, nearly 30% of all domestic consumption was supplied by foreign imports. During that year the Organization of Petroleum Exporting Countries (OPEC) imposed an embargo on crude oil shipments to the United States, causing severe hardships in both the industrial and private sectors. Even though the embargo was short-lived, it did have far reaching consequences, namely: (a) the rapid escalation of fuel prices, and (b) the creation of a nationwide awareness that fuel supplies are very uncertain and subject to instant interruption. In spite of this occurrence, the foreign oil dependency has been allowed to escalate to the point where nearly 50% of the United States' requirements are now imported.

Because of the above fuel situation, there is reason for concern that energy in appropriate quantities may not be available in the future to meet mobilization requirements at the Army Manufacturing and Loading plants. Even if these requirements can be satisfied, it is certain that manufacturing costs will be adversely affected by rapidly escalating fuel prices. To insure that mobilization requirements can be met at an economically acceptable level, it became evident that a comprehensive energy conservation program would have to be established. MMT Project 4281, "Conservation of Energy at Army Ammunition Plants," was established to introduce advanced energy conservation technology into the process operations at munitions plants.

This report describes the design, installation, and test evaluation of a heat pipe waste-heat recovery unit by Grumman Aerospace Corporation for a multi-base propellant forced-air dry house at the US Army Armament Research and Development Command (ARRADCOM) in Dover, New Jersey. Detailed analyses are also furnished on energy/cost savings projections for the installation of similar recovery units at Radford Army Ammunition Plant, a major multi-base propellant manufacturing facility. The work was performed under Contract DAAA21-77-C-0021 as part of Task 4, Energy Recovery from Waste

Heat, of Project 4281. The Special Technology Branch, Manufacturing Technology Division, LCWSL, ARRADCOM was responsible for the assignment of funds and technical direction of the project effort documented here.

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SUMMARY

A prototype waste heat recovery system suitable for propellant forced-air dry house applications has been designed, built, and tested at the US Army Armament Research and Development Command (ARRADCOM) Dover site, under actual field conditions. The system uses a heat pipe heat exchanger (HPHX) that can recover up to 62% of the waste heat from nitroglycerin (NG) contaminated exhaust air at 3000 cfm (5100 m^3/hr) and 140°F (60°C) without compromising safety. A pneumatically actuated intake-air bypass control system is used to prevent the condensation of nitroglycerin in the exhaust stream for any ambient inlet condition by maintaining the exhaust temperature above the NG dew point. The set point is adjustable to accommodate various propellant and air flow combinations.

Three modes of operation are provided: the dry cycle, where the waste heat is recovered from the 140°F (60°C) exhaust stream and used to preheat the ambient intake air; standby, where the dry house is kept at a nominal temperature of 80°F (27°C) awaiting the loading of propellant; and purge, where the dry house is empty, the heat pipe heat exchanger is completely bypassed and 140°F (60°C) air circulates through the system to evaporate any traces of nitroglycerin from dry house and heat exchanger surfaces.

Performance evaluation proved the operational feasibility of the system. In dry cycle operation, an overall energy recovery effectiveness of about 40% was measured for winter operation when the exhaust temperature was limited to 80°F (27°C). An effectiveness value of 62% was estimated when exhaust temperature limits are not imposed and/or the ambient temperature is above 44°F (7°C).

The basic system concept can be used either as a retrofit for existing dry house installations or as an integral part of a more comprehensive resource conservation effort which would include solvent recovery. For the latter application, the condensation and collection of nitroglycerin would result in two additional cost savings: higher energy recovery effectiveness resulting from an unrestricted exhaust temperature and a smaller NG scrubber requirement for the solvent recovery apparatus.

A supporting economic evaluation was performed which assumed Radford Army Ammunition Plant (RAAP) design data and full mobilization. By retrofitting existing dry houses, a return on investment (ROI) of up to 69% can be realized. If used as part of a modernized facility with solvent recovery, the ROI is estimated to be 45%.

INTRODUCTION

As part of an extensive energy conservation effort, the Department of the Army is applying state-of-the-art energy recovery devices to achieve more economical operation of its chemical process facilities. The purpose of this program is to demonstrate a prototype energy recovery system suitable for one of the more obvious sources of waste heat—the multi-base propellant forced-air dry house.

With current operating procedures, outside air is heated by steam coils to 140^{0}F (60^{0}C), then forced through the dry house once. As the warm air circulates through the stacked trays of solvent-wet propellant, it absorbs acetone and ethyl alcohol and picks up traces of nitroglycerin. It is then discharged directly into the atmosphere where the low dew-point trace elements (e.g., nitroglycerin) can condense within an exhaust hood, which covers the exhaust opening, and drip into a catch-pan containing an inerting solution. This is a single-pass process with the exhaust air temperature nearly equal to the warm dry house inlet air temperature. *Until now, no effort has been made to reclaim* this wasted heat, which can amount to 9 or 10 million pounds of steam per year for each dry house during mobilization conditions at a propellant manufacturing facility such as Radford AAP.

By using an air-to-air heat pipe heat exchanger (HPHX) between the exhaust and intake streams, large amounts of this waste heat can be used to preheat the cold intake air. This results in a significant reduction in the energy demands for the propellant drying process and significant cost and energy savings to the Army.

Although there are three other competitive air-to-air energy recovery devices commercially available, the HPHX was selected as the most responsive to the performance requirements of the forced-air dry house application. Rotary regenerative units (i.e., thermal wheels) were eliminated because cross-stream contamination is permitted by their design (rotation of heat transfer surfaces between adjacent air streams). At best, this leakage can amount to 10% of the air flow. In addition, relatively high maintenance is associated with the units' seals, drive, and motor. Plate-fin heat exchangers are generally of cross-flow design, which limits their theoretical effectiveness to 75%

as opposed to 100% for counterflow units. Counterflow designs have more complicated ducting arrangements since both intake and exhaust flows are turned 180°. Since sealing between air streams is required over the entire area of thin sheet-metal heat transfer surfaces, long-term corrosion can also be a problem. Finally, the run-around-loop, which uses an intermediate heat transfer fluid, does not lend itself to this application, being better suited to widely separated air streams where side-by-side ducting would be impractical. It also requires electric power and added maintenance to accommodate the pumped fluid loop.

The performance advantages of the HPHX are summarized below:

1. High effectiveness—up to 80% with simple fin-tube configurations.

2. Simple control over effectiveness—wide variations can be achieved by either mechanically tilting the unit slightly or by diverting a portion of the air flow around the heat exchanger.

3. Separation of inlet and exhaust streams—a sealed center partition prevents cross-stream infiltration and contamination.

4. Easy maintenance and cleaning of heat transfer surfaces—spray nozzles and access ports are readily accommodated.

DRY HOUSE DESIGN DATA

Nominal design data specifications for the forced-air dry house at ARRADCOM are as follows:

Dry house operating temperature	= 140°F (60°C)
Air temperature	= 50.7°F (10.4°C), yearly average
Air flow	= 3000 acfm (5100 m^3/hr)
Dry house capacity	= 3000 lb (1362 kg) propellant

The exhaust gas analysis of M7 propellant is as follows:

Nitroglycerin	= 2.5 ppm max
Alcohol	= 150 ppm max
Acetone	= 150 ppm max

The exhaust gas analysis was obtained on M7 propellant early in the program because M30, the propellant selected for waste heat recovery evaluation tests, was not available at ARRADCOM at that time. M7 has a slightly higher NG concentration than M30; however, the propellant loading level planned for the evaluation tests, 2500 lb (1134 kg) using M30, compared to that used for the M7 exhaust analysis, resulted in approximately the same amount of NG in the dry house. Consequently, the NG concentration in the exhaust using M7 propellant was estimated to be approximately equivalent to that encountered during the evaluation tests. Similar data from RAAP using M30 propellant in larger dry houses using standard drying procedures, 5500 cfm (9350 m^3/hr), showed NG concentration in the exhaust to be 2.6 ppm (max), very similar to that obtained at ARRADCOM using M7.

SYSTEM DESCRIPTION

The forced-air dry house (FAD) waste heat recovery system at ARRADCOM has been designed for three distinct operating modes: dry cycle, standby, and purge.

1. Dry cycle is the primary operating mode when up to 3000 lb (1362 kg) of propellant are forced-air dried for 40 hours. Waste heat is recovered from a 140°F (60°C) exhaust stream and is used to preheat clean intake air. The exhaust stream exit temperature from the heat exchanger is limited to 80°F (27°C) to avoid condensation of nitroglycerin on the heat exchanger and exhaust duct surfaces.

2. During the standby operating mode, the dry house is maintained at a nominal temperature of 80°F (27°C) awaiting the loading of propellant. Waste heat is recovered and the exhaust stream exit temperature is limited to 40°F (4°C) to preclude frost accumulation.

3. The purge operating mode is activated when the dry house is empty. The HPHX is completely bypassed and air at 140°F (60°C) circulates through the system to evaporate all traces of nitroglycerin from dry house and heat exchanger surfaces. This mode would most likely be run before entering an extended dormant period or prior to extensive refurbishment and disassembly of components.

Components

A general layout of the demonstration FAD waste heat recovery system, which was installed in dry house building 1414 at ARRADCOM, Dover, New Jersey, is shown in figure 1. The actual finished facility is shown in figure 2. The heart of the system is the heat pipe heat exchanger, which recovers the waste heat, and the automatic intake-air bypass control system which is designed to prevent the condensation of nitroglycerin in the exhaust stream for any ambient inlet condition.

Heat Pipe Heat Exchanger

The HPHX recovers waste heat from the contaminated exhaust stream and transfers it to the clean intake air stream. The all-aluminum unit contains 180 individually sealed heat pipe tubes that are mechanically press fitted to 12-mil-thick plate fins using conventional heat exchanger design practice. The heat pipe working fluid is Freon-12, chosen because it has a higher than atmospheric vapor pressure at all anticipated service temperatures and is non-reactive with the exhaust stream contaminants in the event of a tube leak.

For this application, a fin spacing of 12 fins per inch (4.7 fins/cm) was considered the closest spacing consistent with reasonable maintenance practices in a clean air stream environment (non-particulate). A standard tube pattern consisting of 0.625-in. (1.59-cm) OD tubes arranged in staggered rows and placed on 1.5-inch (3.8-cm) centers was also judged acceptable. Configuration details are summarized in figure 3, which gives the applicable performance variation at a 3000-cfm (5100-m^3/hr) flow rate as a function of face velocity and number of tube rows.

The final heat exchanger size (i.e., frontal area, number of rows) was determined by selecting the configuration that best combines a large net energy savings and a high rate of return on investment. For instance, using the highest possible effectiveness unit increases exhaust stream energy recovery but requires a larger, more expensive unit with more heat pipes and/or closer finning, which results in larger pressure drops and greatly increased fan power requirements. Increased fan power decreases the net energy savings of the system and, combined with the increased heat exchanger costs, also decreases the rate of return. Based on the results of the ROI analysis (Fig. 4), a final heat exchanger design having a nominal frontal area of 6 ft^2 (0.56m^2) per side and nine rows of heat pipes (20 heat pipes per row) was selected. The nominal heat exchanger effectiveness is 65% for balanced intake and exhaust design flow rates of 3000 cfm (5100 m^3/hr).

Primary Steam Coil

Located in the intake duct behind the heat exchanger, the primary steam coil boosts the intake air temperature to maintain a nominal 140^{o}F (60^{o}C) inlet to the dry house. A no-freeze steam coil, it is designed to automatically drain condensate when the system is inactive. The nominal coil capacity is 300 lb/hr (136 kg/hr) and the supply line pressure is between 5 and 10 psi (34 and 69 kPa). A pressure reducing and flow control station, located in a small anteroom at the back of the dry house, regulates the steam flow to provide the required air temperature.

Dry House Booster Steam Coil

Duct heat losses and other system inadequacies are compensated for by the booster steam coil which boosts intake air temperature and maintains a $140^{\circ}F$ ($60^{\circ}C$) inlet to the dry house. This steam coil, with a nominal 125-lb/hr (57-kg/hr) capacity, was part of the original dry house installation. A cam/follower temperature controller, located in the main fan house, is used to make the final adjustment to the dry house inlet temperature. Air temperatures as high as $176^{\circ}F$ ($80^{\circ}C$) can be achieved by a suitable adjustment of the cam/follower stylus.

Explosion Proof Fan/Motor

The centrifugal fan circulates 3000 cfm (5100 m^3/hr) with a static pressure rise of 3 in. water (745 Pa). The fan is made from non-sparking materials and driven through an adjustable V-belt drive by a 3-hp, 460-volt-3 ø 60-Hz electric motor. The 18.25-in. (46.36-cm) diameter impeller wheel has a nominal speed of 1500 rpm.

The fan is located upstream on the intake side of the heat exchanger so that the air pressure on the intake side exceeds that on the exhaust side. Thus, if any leakage occurs across the center partition, it will be from the clean intake stream to the contaminated exhaust stream and not vice-versa.

Control Systems

The FAD waste heat recovery system has three separate control systems: one to regulate steam flow to the 300-lb (136-kg) coil, one to operate the intake air damper system, and one for the low and high temperature alarms. Although they function independently of one another, their set points must be synchronized to achieve proper system operation.

Steam Flow

The steam flow to the primary 300-lb (136-kg) coil is regulated by two manual valves that are located in the dry house anteroom. One valve (V-1) is used for dry cycle and purge operations and sets the steam coil exhaust air temperature to $140^{\circ}F$ ($60^{\circ}C$). The other valve (V-2) is used for the standby mode to set the exhaust air temperature to $80^{\circ}F$ ($27^{\circ}C$). In both cases a pneumatic signal from a temperature sensor is used to properly modulate a pressure reducing valve which actually regulates the steam flow to the coil. In all control modes, a high limit safety switch shuts down the fan and sounds an alarm if the duct air temperature reaches 180°F (82°C).

Automatic Intake Bypass

This system controls cold side heat exchanger flow with adjustable dampers so that exhaust stream temperature is always kept above the nominal nitroglycerin dew point of 80°F (27°C). The exhaust stream temperature setting is adjustable to accommodate differences in anticipated dew point.

The temperature control system for the heat exchanger consists of a bypass duct over the top of the exchanger intake side, a set of dampers in the bypass duct and the through duct to the exchanger intake side, pneumatic damper actuators, and a pneumatic temperature controller. As the exhaust air temperature approaches the NG dew point, the bypass dampers progressively open while the in-line heat exchanger dampers close. If a system malfunction causes the exhaust temperature to continue to drop, an alarm on the control panel in the fan house sounds and an interlock is established which automatically shuts down the intake fan.

A two-position pneumatic switch (SW1) must be set for the desired operating mode, (dry cycle, purge, or standby). The exhaust stream temperature limits for the automatic dampers can be easily adjusted by changing the thermostat settings for the pneumatic actuators. Two thermostats are located in the dry house anteroom beside the master pneumatic control selector switch (SW1) and a dial thermometer. The thermometer registers the exhaust gas temperature for the damper control system. One thermostat controls the dry cycle and purge temperature limit (T3), and the other, the standby temperature limit (T4). In each case a clockwise turn of the set screw increases the set point temperature. The nominal deadband for each thermostat is about 4°F (2.2°C). That is, a nominal ±2°F (1.1°C) variation can be expected around the set point temperature so that the actual setting should be at least 2°F (1.1°C) above the minimum required exhaust stream temperature.

Alarms

The system contains three alarms:

1. For low temperature limit of heat exchanger hot-stream exhaust for dry cycle and purge operating modes, 75°F (24°C) nominal.

2. For low temperature limit of heat exchanger hot-stream exhaust for standby mode, 37°F (3°C) nominal.

3. For high temperature limit for 300 lb (136 kg) steam coil outlet air, 180°F (82°C).

In all cases, when a temperature limit has been exceeded the electrical power to the fan motor is interrupted, which stops the air flow and causes an alarm to sound. In addition, one of the two fault-indicator lights on the mode select panel (SW2) lights up to indicate whether the high or low temperature limit has been exceeded. The mode select panel is located in the main fan house. A silence button on the mode select panel stops the audible alarm, but leaves the fan power and fault lights unaffected. Correcting the cause of the fault restores the system to its operational state.

The high temperature limit at the steam coil has been set at 180°F (82°C) and has not been designed to be reset conveniently, for safety considerations. However, the low temperature exhaust limits can be readily changed by adjusting either of the temperature controllers mounted on the intake ducting between the HPHX and the primary steam coil. The controller on the right (T2) controls the low temperature limit alarm setting for both the dry cycle and purge operating modes. The controller on the left (T1) controls the low temperature limit for the standby mode. The nominal settings for each are 37°F (3°C) for T1 and 75°F (24°C) for T2.

To avoid sounding the alarms during either an initial start-up from a cold ambient condition or during a changeover from standby to dry cycle or purge, the applicable T1 or T2 controller must first be set below the existing heat exchanger exhaust temperature. Then the required operating mode can be entered without sounding the alarm. The exhaust temperature quickly increases above the desired low temperature limit, after which the alarm controllers can be set to their proper positions without incident.

System Installation

Figures 5 through 11 show a number of views of the recovery system installation at various stages of completion. Significant installation items are identified in all figures.

The most important design features of the FAD waste heat recovery system from both operational and safety considerations are summarized below:

Heat Pipe Heat Exchanger

1. Uses high vapor pressure fluid (Freon-12)

2. Non-sparking, aluminum materials for HX tube/fin combination

3. No condensation or freezing of nitroglycerin vapor permitted on HX surfaces

System

1. Automatic pneumatic-type inlet air bypass control to maintain exhaust gas temperature above NG condensation point

2. Primary steam coil with an automatic control valve and safety shutoff to provide a constant 140°F (60°C) dry house ambient temperature during normal operation and insure a 180°F (82°C) maximum air temperature limit

3. Explosion-proof construction of all electrical components in both intake and exhaust stream duct work. Installation per National Electric Code (NEC) Group D atmosphere and Class I, Division 2, location

4. Non-sparking materials for contaminated exhaust stream

5. Push-through fan location so that static pressure of intake stream exceeds that of exhaust stream

Ducting

1. Aluminum one-piece duct on exhaust side (smooth surfaces, no cracks)

2. Silicone caulking (non-NG absorbing) on all attachment flanges and seams

3. Aluminum attachment hardware

4. Galvanized ducting suitable for intake side only

5. Simple aluminum exhaust hood similar to existing configuration (nitroglycerin condensate collects in drip pan containing an inerting solution)

6. 2-inch-thick fiberglass PF board insulation

Instrumentation

1. No flow measurement on exhaust side

2. Intrinsic safety thermocouples per NEC

System Operation

The control settings for proper system operation are summarized in table 1. In the case of the Building 1414 prototype installation, it is assumed that air flow to the main fan house has been blocked by closing the appropriate damper and that the old dry house wall vent has been properly sealed. These precautions would not be needed for most actual field installations since all traces of the old system would be removed.

The applicable alarm controller T1 or T2 must be overridden during either an initial start-up from a cold ambient or during a change over from standby to dry cycle or purge operation. As soon as the exhaust temperature exceeds the desired minimum alarm set point, the controllers can be reset to this value.

A higher booster coil temperature setting of 158°F (70°C) is recommended during purge operation to compensate for tare losses through the heat exchanger. This is discussed in detail in the test evaluation section.

Miscellaneous Comments

- Although the dry house building was refurbished prior to the installation of the waste heat recovery system, the double loading doors and frame were not modified, which left fairly sizable air gaps when the doors were closed. The total vent area was estimated to be 43 in.^{2} (279 cm^2). During the initial check-out runs, the high inside static pressure, $\approx$ 0.5 in. of water (124 Pa) resulted in an excessive amount of air leakage from the door gaps and decreased air flow through the heat exchanger exhaust side. To remedy the situation and provide exhaust air flow closer to the 3000-cfm (5100-m^3/hr) design point, a curtain of conductive polyethylene plastic sheet (Velostat) was draped across the inside of the dry house doors. Thus the internal positive pressure helped to press the plastic sheets against the doors and thereby seal the air gaps.

- The primary 300-lb (136-kg) steam coil is designed to drain all condensate from its tubes, preventing damage due to freezing when the unit is not operating. However, the steam lines which supply the coil are susceptible to freezing damage and must be drained whenever the steam flow to the primary coil is interrupted for long periods during subfreezing weather.

HAZARDS ANALYSIS/SAFETY

To insure that the FAD waste heat recovery system met U.S. Army hazards and safety criteria, the Allegany Ballistics Laboratory, a division of Hercules Incorporated, was retained as an independent hazards consultant. They established general safety guidelines within which the system design was formulated, and evaluated the final installation. The basic objectives were achieved: to design a system which uses conventional heat exchanger and duct designs, and to fabricate and install the FAD system with no sacrifice in safety.

Two separate hazards analyses were provided in accordance with the requirements of ARMCOM Regulation 385-4. The details are well documented in both a preliminary hazards analysis report (Ref. 1) and a final hazards analysis report (Ref. 2). The preliminary hazards analysis provided design and operating guidelines during the design phase of the project. Basically, these guidelines were predicated on avoiding the hazards associated with the condensation and possible freezing of nitroglycerin liquid in the duct system and on the heat exchanger. The specific design features which helped to accomplish this objective have been summarized in the previous sections. More detailed discussions can be found in the referenced preliminary hazards analysis report.

The final hazards analysis report provides a quantitative system analysis of the design which determines the failure mode and evaluates incident probability for the hazards previously identified in the Preliminary Hazards Analysis. Of the 16 potential hazards evaluated, two violated the ARMCOMR 385-4 criterion of incident probabilities greater than 1.5×10^{-2} per week at a 95% confidence level. Both were Category III (critical) designations. One dealt with friction initiation of NG on the exhaust hood (probability of 1.2×10^{-1}) and the other with initiation when the access doors to the air intake plenum are removed for replacement of filters (probability of 1.8×10^{-2}). The primary recommendations for preventing these hazards are as follows:

1. Install an air deflector between the exhaust duct at the heat exchanger and the line-of-sight to the intake plenum.

2. Do not remove exhaust hood to inspect HX.

3. Since high NG dew points could be encountered due to uncertainties in NG vapor pressure data, raise exhaust outlet limit to at least 85°F (29.4°C) and alarm setting of 80°F (26.7°C).

4. Change galvanized steel access doors on intake ducting and fan plenum to plastic doors.

Nitroglycerin (NG) Contamination

Nitroglycerin condensation and subsequent contamination of the exhaust side surfaces can be prevented by maintaining the exhaust air at a higher temperature than the nitroglycerin dew point. The NG dew point of the exhaust air is dependent on the NG vapor concentration in the air. The vapor concentration is in turn a function of the air flow rate through the dry house, the dry house temperature, the type of propellant being dried, and the propellant in the house. Figure 12 presents a plot of dew point versus vapor concentration based on NG vapor pressure determinations by Crater (Ref. 3). Crater's data, which is the latest available, is presented in figure 13. It should be noted that other vapor pressure data found in the literature (Ref. 4) give significantly higher dew points than those reported in figure 13. However, these earlier data are not considered a accurate as more recent findings.

Typical maximum NG vapor concentrations measured in ARRADCOM and Radford AAP dry house exhaust ducts are shown in table 2. For 140°F (60°C) air flows in the range of 3000 to 5500 cfm (5100 to 9350 m^3/hr) with M7 and M30 propellant, the NG dew point of the exhaust air, according to Crater's data, should be approximately 75°F (23.9°C).

Preheating the dry house and exhaust ducting to a temperature above the NG dew point is not necessary before loading the house with propellant. Available RAAP data indicate that for the first three to four hours of the dry cycle the NG vapor concentration in the exhaust, for 7344 lb (3330 kg) of M30 propellant at a 5500 cfm (9350 m^3/hr) air flow, is less than 0.1 ppm. This concentration gives a dew point less than 14°F (-10°C) (Fig. 12). When hot air is introduced to the house at start-up, the exhaust duct temperature rises rapidly above 14°F (-10°C) regardless of the ambient temperature if the exchanger temperature control system is functioning properly.

The most logical causes of NG condensation and contamination on the exhaust side surfaces are: (1) a lower NG vapor pressure than expected (i.e., higher NG dew point); (2) a malfunctioning intake air bypass control system resulting in exhaust temperatures dropping below the NG dew point; or (3) an NG vapor concentration in the exhaust air exceeding 2.5 ppm. Potential causes for higher vapor concentrations in the exhaust are propellant overload, high air-flow rates, and "bootstrapping" resulting from NG contaminated intake air. To illustrate this latter point, consider an NG vapor concentration of 0.5 ppm developing at the intake due to inadequate dispersion of the exhaust gases. If this intake concentration results in a 3-ppm level in the exhaust, which in turn causes an increase in the intake concentration, a steadily increasing exhaust content will ensue. Such a bootstrapping process is limited by the dew point of ambient air. That is, if the exhaust side of the heat exchanger is being controlled at 80°F (27°C) (and the relative humidity is assumed to be zero percent), the NG concentration at the intake hood must eventually build to 0.9 ppm to give condensation in the exchange. As Fig. 12 indicates, this is only possible for ambient air temperatures exceeding 60°F (16°C).

In the normal course of system operation, the exhaust surfaces and intake air plenum should be checked for NG contamination by wiping each surface with a clean cotton swab followed by a chemical analysis. If condensate and contamination occurs due to the NG dew point's being higher than predicted, any of the following steps can be taken to eliminate it:

1. Increase exhaust side operating temperature.
2. Decrease air flow through the dry house.
3. Decrease propellant loading in the house.
4. Decrease input air temperature to the house.

As an added precaution, air filters should be replaced frequently to minimize NG absorption. The recommended analysis of the filters for NG content should help determine how long the filters can safely remain in service without picking up excessive NG. Excessive NG levels can be determined by testing contaminated filters with flame and cap initiation. After removal, filters should be handled as contaminated scrap.

From the standpoint of future installations, the recommended solution to the potential problem of drawing NG condensate droplets into the intake duct is to install secondary air filters external to the duct under the intake weather hoods. The filters should be attached to the duct with nonmetallic hardware without screw threads. Also, the dry house exhaust and intake ducts should be oriented so that, relative to the prevailing wind direction, the intake is upwind of the exhaust. Calculations performed for this study indicate that it is impractical to prevent NG droplets from reaching the intake by increasing the exhaust/intake separation.

Decontamination/Maintenance Procedures

At cold ambient temperatures, nitroglycerin condenses at the point where the exhaust duct discharges to the atmosphere. Since this condensation cannot, as a practical matter, be eliminated, the exhaust duct discharge is equipped with a deflector and drip pan arrangement. The deflector is a single piece, non-sparking, metal awning which deflects the air flow down onto the drip pan under the awning. The drip pan is sufficiently large to catch any condensate dripping off the awning and should be partially filled with a low volatility, low freezing point, NG soluble oil, such as diethylphthalate (DEP), dioctylphthalate (DOP), triacetin (TA), or motor oil. It should be protected from the elements by the awning or a separate overhanging roof.

The drip pan material should be non-sparking and designed to dissipate static electricity. Aluminum or conductive plastic, suitably grounded, are acceptable.

The recommended general decontamination and maintenance procedure for the exhaust ducting and heat exchanger is as follows:

1. Purge the dry house with $140^{o}F$ ($60^{o}C$) air for a sufficient period of time to evaporate any NG which may have accumulated at the tube-fin interface at the heat exchanger. Nitroglycerin evaporation rates are given in figure 14 as a function of the bulk air side film heat transfer coefficient for selected values of diffusivity. For a dilute NG air mixture of $140^{o}F$ ($60^{o}C$), the nominal value of diffusivity is 0.25 ft^2/hr (0.065 cm^2/sec), with a scatter of ± 30% (see Ref. 5). Assuming a worst case, clean uncontaminated exhaust air stream during the purge mode, the net NG evaporation rate is 0.10 lb/ft^2 hr (4.97 mg/cm^2 hr) for a driving vapor pressure differential of 0.018 in. (0.046 mm) Hg. Thus under these conditions,

a 12-mil-deep crack (this is the heat exchanger fin thickness) requires about 12.6 hr of purge time.[1]

2. Disassemble the exhaust duct starting with the deflector hood and drip pan. Prior to removal, wash the hood thoroughly with an NG solvent (i.e., acetone, triacetin, methanol, ethyl acetate, or dichloroethylene) or warm water and detergent to remove any NG accumulations. Be sure that both personnel handling solvent and solvent containers are adequately grounded as these solvents are extremely flammable. Also, maintain ground continuity in the duct sections as they are washed down. Apply NG killer to the hood, particularly in the caulked joint areas, and allow to stand for several minutes. A typical formula for NG killer is given in table 3. CAUTION: Do not apply NG killer until the duct section has been thoroughly washed with solvent or detergent, since contact of killer with any appreciable quantity of NG can result in an explosion.

3. Wash down the exhaust side of the heat exchanger with solvent and spray with killer. Remove the exchanger. (Here it is assumed that the intake duct system has previously been disassembled.)

4. Wash down the exhaust duct sections upstream of the exchanger and disassemble section by section.

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1. The purge cycle mentioned here does not meet conventional standards for decontamination of a part prior to hot work or transportation to non-explosive areas of a plant. In the case of equipment potentially contaminated with NG, decontamination is achieved only after exposure to a minimum of 450°F (232°C) for a minimum of four hours. Further, the 450°F (232°C) temperature must be measured on the most massive section of the part being baked.

TEST EVALUATION

The main test objective was to obtain a quantitative comparison of the total energy consumption for a typical forced-air dry house operating with and without a waste heat recovery system; specifically, a system which uses a heat pipe heat exchanger (HPHX) as the energy recovery device. Additionally, data were obtained on the operation of an automatic intake bypass control system so that its performance benefits could be weighed against the increased system complexity and cost.

Test Description

Three basic system operating modes were evaluated: standby, dry cycle, and purge. During standby, when the dry house is empty and awaiting a propellant load, the dry house air temperature was kept at 80°F (27°C) and the heat exchanger exhaust was limited to 40°F (4°C) to preclude frost accumulation. The dry cycle sequence was representative of that used for M30 propellant at the Radford, VA, Army Ammunition Plant. It consisted of an 11-hour warmup from a standby condition at an average temperature rise of 5°F (2.8°C) per hour, followed by a 40-hour drying cycle at 140°F (60°C) during which the heat exchanger hot-side exhaust temperature was limited to 80°F (27°C). The M30 propellant load for each dry cycle was 2500 lb (1135 kg). After two dry cycles, a 48-hour purge cycle was run to determine the effect of purge time on the presence of nitroglycerin on the heat exchanger surfaces. During the purge mode, the dry house was kept at 140°F (60°C) with the heat exchanger intake completely bypassed. This minimized the heat extracted from the hot exhaust stream, which kept exhaust side surface temperatures relatively warm so that any traces of nitroglycerin could be evaporated.

At the end of each test sequence and at selected time intervals during the purge mode, nitroglycerin contamination was checked at four locations: the heat exchanger hot-side exhaust surfaces and three places on the intake plenum of the fan (e.g., intake hood protective screen, access panel doors, and intake air filters). The test timeline is summarized in table 4.

Test Set-up/Instrumentation

For each test sequence, the energy required to operate the dry house was determined by monitoring the total air stream energy input. This was obtained by measuring the differential temperatures and air flow associated with each heat transfer component (heat exchanger, steam coils).

A system schematic which shows the instrumentation locations is given on figure 15. The data measurements are further defined in table 5, and the principle test equipment used for the measurements is described in table 6.

To satisfy safety requirements, the only instrumentation permitted in the NG-contaminated exhaust stream were the thermocouples before and after the heat pipe heat exchanger (T8, T9). As with other temperature measurements, type-T (copper-constantan) thermocouples of intrinsically safe design were used. For each thermocouple, the wires and measuring junction were enclosed in a sealed stainless steel sheath and grounded to the sheath tip. The electrical signal generated was less than 10 millivolts.

During the final system checkout, the temperature distribution across the face of the heat exchanger hot-side exhaust stream was manually probed to insure uniformity, especially between the alarm and damper control sensor locations and T9 of the data recorder.

The volumetric flow rate of air entering the dry house was obtained by measuring the air velocity at the centerline of the inlet duct, about 20 ft (6 m) downstream of the nearest bend. A careful probe of the duct cross-section determined that the measured centerline velocity was very close to the average duct velocity, giving a difference of less than 0.5%. Thus, the measured value was used directly in all subsequent calculations. The product of the centerline duct velocity and duct cross-sectional area, 2.88 ft^2 (0.268 m^2), gives the volumetric flow rate, cfm(m^3/hr). This value multiplied by the local air density gives the intake mass flow rate. A curtain of conductive polyethylene plastic sheet (Velostat) draped across the inside of the dry house doors minimized the amount of air leakage so that exhaust flow nearly equaled inlet flow.

A kilowatt-hour meter, part of the fan electrical installation, was used to measure HPHX fan power consumption. The total reading was manually recorded prior to and immediately after each data-taking period.

Except for the fan power consumption, all data were automatically recorded at regular time intervals (typically 20 minutes) by a multi-channel data logger which produced a permanent paper tape record of all transducer measurements. The recorder was remotely located in the old fan house which was situated about 100 ft (30 m) from the dry house. A sample of the data record is shown in figure 16. The time is indicated in the lower left corner, hr-min-sec. Channel 000 registers the local fan house temperature, and channels 001 through 011 register the system temperatures as given by the schematic in figure 15. The average duct velocity (in ft/min) is obtained by multiplying the channel 020 reading by 1000. Channel 021 multiplied by 2.0 gives the fan static pressure rise in inches of water.

Test Results

The raw test data for all test sequences were automatically recorded by a multi-channel data logger. Reduced data in the form of transient (hourly) data plots, average values of the significant test parameters, and the overall system energy balance and energy recovery effectiveness for each test sequence are given below. The general procedures used in calculating performance are given in Appendix A.

Certain data have been plotted at 20- or 30-minute intervals (as applicable) to more accurately depict the transient behavior at a particular time (see figures 17-28). This is especially true when observing the outlet temperature of a modulating steam coil (T5 or T7 in Figs. 19, 22, 25, and 28) or, during dry cycle operations, when observing the mixed cold side heat exchanger exhaust temperature (T4 in Figs. 21, 22, 24, and 25) when the intake air bypass dampers are rapidly changing position. The latter seems to occur most frequently with decreasing ambient temperatures in an environment below about $23^{o}F$ ($-5^{o}C$).

The performance for each test sequence is presented by transient data plots of the following parameters:

Type of data	Parameters plotted
1. Overall system performance	Air flow rate, Fan ΔP, T3 (dry house) T10 (ambient)
2. Heat exchanger performance	Cold side: inlet (T1), mixed outlet (T4) Hot side: inlet (T8), outlet (T9)
3. Steam coil performance	Primary coil: inlet (T4), outlet (T5) Booster coil: inlet (T6), outlet (T7)

Standby Mode

The standby operating mode was run for a continuous 67-hour period, before the dry house was loaded with propellant. Data printout intervals were 30 minutes. Figures 17 through 19 illustrate the standby performance for the first 48 hours of the test sequence. The system energy balance for this period is summarized in table 7.

As seen in figure 17, the overall system performance plot, the dry house temperature fluctuated between 75 and 82^{o}F (24 and 28^{o}C) with an average of 78.4^{o}F (25.8^{o}C), while the ambient temperature was generally below 20^{o}F (− 7^{o}C) for an average of 17.6^{o}F (− 8^{o}C). The fan static-pressure rise is relatively steady at about 3 inches of water (744 Pa), but the air flow rate fluctuates more markedly than any of the other runs. The minimum and maximum flow rates vary by about 600 cfm (1020 m^3/hr) from an average value of 3022 cfm (5135 m^3/hr).

All during the standby operation, the hot side exhaust temperature (T9) was kept above the minimum 40^{o}F (4.4^{o}C) set point and varied between 41^{o}F (5^{o}C) and 49^{o}F (9.4^{o}C) (see Fig. 18). The average value was 45.1^{o}F (7.3^{o}C). The temperature drop between the dry house and the hot side heat exchanger inlet was less than 1^{o}F (0.5^{o}C) which was typical of all the test sequences. Energy recovery is evidenced by the transfer of heat between the hot exhaust stream and the cold intake stream. The hot stream experiences a drop in temperature across the heat exchanger (heat loss) while the intake stream temperature increases (heat gain). Ideally, the heat loss and heat gain should balance, and the temperature differences across each stream should be equal for equal mass flow rates. As seen, this is not the case for the standby mode and the disparity between temperature differences is the largest of all the test sequences. One possible explanation is excessive air leakage from the dry house which would result in a much lower exhaust side flow rate and, consequently, a much larger temperature drop for the same cold side heat transfer. Since the ratio of exhaust flow to intake flow is given by the ratio of temperature differences, (T4−T1)/(T8−T9), a flow ratio of 0.45 is indicated, based on average temperature values. This implies that about half the flow is lost, which contradicts both qualitative physical evidence and the results from other test sequences which show more balanced flow streams (flow ratios of 0.76 for dry cycle and 0.84 for purge). In all cases, a plastic curtain was draped across the inside of the loading doors, from ceiling to floor, so that the positive inside pressure would tend to effectively seal all of the edge leaks. This procedure was necessitated because originally the leakage was large due to warped and ill-fitting door to door-jam surfaces. In addition to poorly sealed doors, another possible explanation for such a large flow imbalance is that the original exhaust hood opening was not sealed shut. But this seems unlikely, since it was covered on the inside by an aluminum sheet that was taped along the edges and held in place by an empty 5-gallon drum.

Another possible explanation for the difference in temperature drops between the intake and exhaust streams is the entrainment of water droplets or ice crystals by the intake air stream. If this were the case, then a portion of the recovered waste heat would be lost to the latent heat of vaporization of the water and the capacity to increase the intake air temperature would be proportionately decreased. This can be seen by including the latent heat term in the heat exchanger energy balance, which results in the following expression:

$$(T4\text{-}T10) = \left(\frac{M_x}{M_T}\right)(T8\text{-}T9) - \frac{\lambda}{C_P}\ \bar{\gamma}_\omega$$

where

$(T4\text{-}T10)$ = measured intake stream temperature increase

$(T8\text{-}T9)$ = measured exhaust stream temperature decrease

$\left(\frac{M_X}{M_T}\right)$ = exhaust flow to intake flow ratio

λ = latent heat of phase transformation for water

C_P = specific heat

$\bar{\gamma}_\omega$ = effective specific water content, lb water/lb air

Assuming a flow ratio of 0.80 (the average value measured for the dry cycle and purge modes), a specific water content of 0.0023 lb water/lb air is calculated for the measured temperature differences during the standby test. This is equivalent to about 3.7 gallons/hr, which seems high.

A possible source of this water could have been condensate formation at the primary steam coil drain which was located near the intake air plenum.

This presented a problem during system checkout runs by causing moisture to be drawn into the air intake system, forming ice on the filters and producing a high pressure drop until a separator was placed between the drain and the air plenum.

The total energy recovered by the heat exchanger was calculated for both intake and exhaust stream temperature differences, using the average intake stream flow rate of 3022 cfm (5140 m^3/hr). The energy recovery during standby is 2.407×10^6 Btu (705 kw-hr) from intake conditions and 4.949×10^6 Btu (1449.6 kw-hr) from exhaust conditions. The average value, 3.678×10^6 Btu (1077.3 kw-hr), represents an overall energy recovery effectiveness of 39% when ratioed to the calculated total energy requirement of 9.345×10^6 Btu (2737.2 kw-hr). Using the lower intake stream value in the ratio yields an effectiveness of only 26%.

The transient behavior of the steam coils is shown in figure 19, where the primary coil (T5) does most of the modulating. This rapid "spiking" makes the determination of steam coil heat transfer somewhat uncertain. Generally, because of more stable temperatures, it would be more accurate to calculate the total energy requirement based on dry house and ambient temperature differences (T3-T10), then subtract the calculated intake stream heat exchanger energy recovery which is based on temperatures (T4 and T10).

Dry Cycle No. 1

Transient performance data for dry cycle No. 1 are given in figures 20 through 22. The system energy balance for this cycle, consisting of an 11-hour warmup followed by a 40-hour drying period, is summarized in table 8. The 11-hour warmup, which was regulated by manual adjustments to the booster coil, was fairly well controlled to give a rise rate of about 5°F (2.8°C) per hour. During the 40-hour dry cycle, the dry house temperature averaged 140.7°F (60.4°C), and the intake air flow rate was stable at an average value of 3013 cfm (5120 m^3/hr). This run proved the most taxing on the automatic air bypass control system because the combination of low, below about 23°F (-5°C), and rapidly decreasing ambient temperature caused very rapid modulation of the intake/bypass dampers, which is evidenced by the high frequency spikes recorded for T4 (Fig. 21) from 42 to 46 hours. Corresponding rapid modulation of the primary steam coil outlet temperature, T5, is shown in figure 22. In spite of these variations, the exhaust side outlet temperature, T9, stayed above the 80°F (27°C) set point and averaged 85.9°F (29.9°C) for the entire run. As shown in figure 21, the exhaust stream temperature drop across the heat exchanger more nearly equals the temperature rise across the intake stream, which implies a much more balanced flow situation than previously experienced during the standby mode. An exhaust-to-intake air flow ratio of 0.76 is calculated on average temperature values.

Dry Cycle No. 2

As shown in figures 23 and 25, the system performance for the second dry cycle is similar to the first. The system energy balance for dry cycle No. 2 (11-hour warmup; 40-hour drying cycle) is summarized in table 9. The average ambient temperature was somewhat warmer [25.8°F (-3.4°C) vs 23.4°F (-4.8°C)] , and the average air flow rate slightly lower 2915 cfm (4953 m^3/hr) vs 3013 cfm (5120 m^3/hr)] . Large drops in the mixed intake temperature (T4), which signify complete bypass of the heat exchanger, are also noted with cold and decreasing ambient temperature. The flow balance between intake and exhaust is also similar to the first dry cycle, having an average ratio of exhaust-to-intake air flow of 0.76.

Purge

Figures 26 through 28 reflect the transient performance data for the 48-hour purge test sequence. The system energy balance for this phase of the evaluation is summarized in table 10.

During the purge test sequence, the dry house was kept at an average temperature of 140.6°F (60.3°C) with an average air flow rate of 2990 cfm (5081 m^3/hr). The decreased value of fan pressure rise compared to dry cycle runs, 2.6 vs 2.9 in. of water (645 vs 719 Pa), is due primarily to the elimination of the pressure drop through the intake side of the heat exchanger.

Although the intake side of the heat exchanger was fully bypassed, there was a measurable amount of heat transfer between exhaust and intake air streams. This was probably a combination of the following: leakage through the closed damper blades; convective eddy currents at the outlet side of the heat exchanger intake stream caused by backwash from the over-the-top bypass stream; and forced-convection heat transfer from the heat exchanger frame surfaces, contacting the bypass air stream. The net result is an exhaust stream outlet temperature that averages 121.4°F (49.7°C), about 19.6°F

(10.3°C) below the desired 140°F (60°C) uniform purge temperature. There are two possible solutions to this problem: either the purge time can be extended as necessary to insure complete evaporation of trace nitroglycerin; or the dry house temperature can be increased by a compensating amount so that the exhaust exit temperature is near 140°F (60°C). At the lower 122°F (50°C) exhaust temperature, the NG vapor pressure is about one-half the value at 140°F (60°C), 0.00087 vs 0.0018 in. (0.022 vs 0.046mm) of Hg (see Fig. 13). This means that the evaporation rate will decrease proportionately so that the required purge time will approximately double. The calculated exhaust to intake flow ratio for the purge mode is 0.84.

Summary of System Performance

The significant parameters used in the performance calculations for all system operating modes are summarized in table 11. These include pertinent recording tape data (print interval, number of data points) and the summations of velocity and selected system temperatures. The average values for these significant test parameters are given in table 12. Finally, the overall system energy balance, as determined by the procedures outlined in Appendix A, is presented in table 13 for all test sequences.

It should be emphasized that the lower temperature limit placed on the exhaust side outlet prevents NG condensation but also restricts the effectiveness of the energy recovery system during operation at low ambient temperatures. The potential to recover greater amounts of energy can be more fully realized by either lowering the exhaust side temperature limit or operating the system at warmer ambient temperatures. Then the overall energy recovery effectiveness could approach the 65% design point.

The effect of ambient temperature on energy recovery can be observed by comparing the energy recovery data from the two dry cycles. The second dry cycle test had a warmer average ambient, 25.8 vs 23.4°F (-3.4 vs -4.8°C) and also showed values of effectiveness about two percentage points higher. This effect can be further illustrated by comparing instantaneous data for exhaust side effectiveness at the coldest and warmest ambient conditions for both dry cycles.

Date	Time	T8	T9	T1	T10	$\eta = \frac{T8-T9}{T8-T10}$
		Temperature, °F (°C)				
Dry cycle No. 1						
3-1-78	0640	136.9(58.3)	82.1(27.8)	17.4(-8.1)	15.7(-9.1)	0.452
2-29-78	1440	135.1(57.3)	82.0(27.8)	37.2(2.9)	32.0(0.0)	0.515
Dry cycle No. 2						
3-8-78	1200	137.7(58.7)	81.8(27.7)	35.3(1.8)	25.8(-3.4)	0.500
3-9-78	1300	144.4(62.4)	82.5(28.1)	44.4(6.9)	34.7(1.5)	0.564

The exhaust side effectiveness increases with increasing ambient temperature (T10). By extrapolating the data, an energy recovery effectiveness of 62% is indicated for ambient temperatures above 45°F (7.2°C).

Operation

During the checkout runs, an appreciable amount of ice accumulation was noted on the fan intake plenum screening and air filters when the system was run over night during cold ≈20°F (-6.6°C) ambient temperatures. Large increases in fan static pressure rise resulted. The cause of the ice formation was the close proximity of the primary steam coil condensate drain to the plenum air intake openings. This resulted in an excessive amount of moisture ingestion and subsequent freezing. A baffle system which redirected the condensate cloud solved the problem. Similar precautions should be taken for future installations.

It was necessary to manually override the exhaust temperature alarm system (i.e., lower the set point) during startup from a dormant state and during changeover from standby to dry cycle or purge operation. Otherwise, the instantaneous exhaust air temperature would have registered below the required set points, 40°F (4.4°C) for standby and 80°F (27°C) for dry cycle, the alarm would have sounded, and power to the fan motor stopped. The manual override permitted acceptable exhaust air temperatures to be reached in a matter of minutes, at which time the alarms were reset to their proper positions. Future systems should incorporate an automatic override to preclude unintentional alarm activation during startup and changeover.

Results of the qualitative nitroglycerin sensitivity tests which were made after the designated operating modes are summarized in table 14. The generally positive results at the fan plenum location warrant quantitative testing before final recommendations can be made. However, at worst they indicate that there must be more separation and/or baffling between the heat exchanger exhaust and the intake plenum openings and/or regularly scheduled treatments with NG killing solvents.

After the second dry cycle test, a stalagmite of unknown composition was noticed on the ground near the support structure at the outside of the heat exchanger exhaust face. This location is beneath the lowest point of the heat exchanger frame which was purposely tilted so that the end of the hot exhaust stream would be about 3 in. (7.6 cm) below the end of the colder intake stream. This gradient insures that the heat pipes operate in a reflux boiler mode, as designed.

A quantitative laboratory analysis of the stalagmite showed it to be 99% pure nitroglycerin. Inspection of the heat exchanger exhaust, support frame, and drip pan revealed two deficiencies: the support structure was positioned too close to the heat exchanger exhaust face; and the heat exchanger frame did not overhang the drip pan by a sufficient margin. Thus, NG condensate was allowed to run along the heat exchanger frame, drip onto the support frame, then onto the ground where it solidified in the cold air. This situation can be remedied in the ARRADCOM prototype system by installing an aluminum (or plastic) deflector shield to direct NG condensate from the heat exchanger directly into the drip pan containing the NG inerting fluid.

The fact that there was nitroglycerin condensate at all on the heat exchanger also indicates that the NG dew point is higher than predicted in reference 3. A condensate-free operation can be achieved by increasing the exhaust temperature set point to perhaps 90°F (32°C). This action, of course, would further curtail potential energy recovery by decreasing the overall effectiveness of the system at all ambient temperatures below about 57°F (14°C).

ECONOMIC EVALUATION

The following economic evaluation of FAD waste heat recovery systems at a representative US Army Ammunition Plant, Radford AAP, conforms to the requirements of ARMCOM Pamphlet 37-2.

The existing Radford installation consists of 24 dry houses, each with four separate rectangular bay areas. Each bay receives 5500 acfm (9350 m^3/hr) of 140°F (60°C) air. Two air moving systems (fan houses), located on opposite sides of each dry house, supply two bay areas. One part of the economic analysis for waste heat recovery at these facilities considers two basic configurations: separate 5500-acfm (9350-m^3/hr systems for each bay area, and one 11,000-acfm (18,700-m^3/hr) system for every two bay areas. Each configuration has two options: without NG condensation, which implies automatic controls, and with NG condensation.

The Army, as part of its modernization and resource conservation programs, is also considering building new forced-air dry house facilities which incorporate solvent recovery systems. The projected air flow for these new systems is 2000 acfm (3400 m^3/hr). Waste heat recovery is also being considered for these new dry houses. In this instance, permitting NG condensation at the heat exchanger would be especially desirable; energy recovery effectiveness would be maximized with a possible further benefit coming from a lessening of the condensing requirements of the NG scrubber, a basic component of the solvent recovery system. Therefore, a second part of the economic analysis of the waste heat recovery system considers two additional operating configurations: one fan per bay, 2000 acfm (3400 m^3/hr) each, or one fan per two bays, 4000 acfm (6800 m^3/hr). Both configurations will permit NG condensation at the heat exchanger. A system designed to permit NG condensation would essentially eliminate the need for automatic controls. This option would have an additional benefit of lower equipment and installation costs (by about $6000/unit, non-discounted).

The following system and cost assumptions were applied to the economic analysis:

1. Mobilization conditions:

   Yearly dry cycle = 7595 hours; 86.7% duty cycle

   Yearly standby operation = 1165 hours; 13.7% duty cycle

Recovery savings from standby operation are neglected in this analysis. If this approach were adopted, additional savings of approximately \$25,000/year (existing dry houses) or \$9200/year (modernized dry houses) would be realized (1980 estimate) which would be expected to increase by 5%/year.

Purge neglected

2. Dry house conditions:

Air inlet temperature - dry cycle = $140^{o}F$ ($60^{o}C$)

- standby cycle = $80^{o}F$ ($27^{o}C$)

Air flow rate - existing = 5500 acfm (9350 m^3/hr)

- modernization = 2000 acfm (3400 m^3/hr)

Average ambient air temperature = $53^{o}F$ ($12^{o}C$)

3. Energy recovery effectiveness (n):

For auto control systems - dry cycle, n = 0.40 for three coldest months

n = 0.62 for nine warmest months

- standby, n = 0.30

For condensing systems - n = 0.62

4. Electrical power consumption [referenced to 3000 acfm (5100 m^3/hr).]

Dry cycle = 0.8 kw-hr per hr

Standby cycle = 2.4 kw-hr per hr

Assuming that the air velocities through the ducts, 1000 fpm (5 m/sec), and heat exchanger, 500 fpm (2.5 m/sec), remain the same (i.e., constant static pressure requirement), power consumption of various size systems will vary directly as the flow and will be determined accordingly.

5. Cost data:

Steam = \$3.80 per 1000 lb (1977)

Electricity = \$0.02531 per kw-hr (1977)

Cost base year is 1979

Coal fuel costs increase by 5% per year (in accordance with guidance from DARCOM Supplement 1 to AR 11-27, dated 27 July 1977).

6. Heat recovery system (unit costs, including 20% quantity discount):

Without auto controls:

2000 cfm (3400 m^3/hr)	= \$16,900
3000 cfm (5100 m^3/hr)	= \$18,400 (equivalent ARRADCOM system)
4000 cfm (6800 m^3/hr)	= \$19,900
5500 cfm (9350 m^3/hr)	= \$24,250
11,000 cfm (18,700 m^3/hr)	= \$35,400

Automatic bypass controls = \$4,800 per unit

Ten year savings economic life after first year in[illegible]tment

No salvage value

A detailed summary of the economic analysis of the options considered is given in tables 15 through 20. Appendix B contains an example which details the procedures used in the analysis. A comparison in terms of savings/investment ratio (S/I) and return on investment (ROI) is presented below.[2]

Existing Dry House

	S/I	ROI%
1. One system per bay, 55[illegible] cfm (9350 m^3/hr)		
No NG condensation (auto controls)	2.68	38.8
With NG condensation	3.52	51.1
2. One system per two bays, 11,000 cfm (18,700 m^3/hr)		
No NG condensation (auto controls)	3.87	56.1
With NG condensation	4.82	69.3

2. The savings/investment ratio is calculated at the current standard DOD discount rate of 10%. It is defined as the total discounted annual operating savings over a 10-year period, divided by the discounted non-recurring costs, which are assumed to be incurred before the first savings period. When this 10% discount rate produces a S/I ratio of 1.0, the value of return can be stated as being 10%. For any other S/I ratio obtained, the actual rate of return would be either more (S/I >1) or less (S/I<1) than 10%. The actual rate of return on investment is determined by finding the interest rate that equates the present value of expected future net savings to the present value costs of the investment outlay.

Modernized Dry House

	S/I	ROI%
1. One system per bay, 2000 cfm (3400 m^3/hr) With NG condensation	1.84	25.6
2. One system per two bays, 4000 cfm (6800 m^3/hr) With NG condensation	3.13	45.4

From an economic viewpoint, the most beneficial configurations permit nitroglycerin condensation and provide one heat recovery system for every two bays.

CONCLUSIONS

1. This system hardware demonstration has proven both the technical and economic feasibility of using heat pipe heat exchanger technology to recover waste heat from propellant forced-air dry houses without compromising safety.

2. During full mobilization, the use of waste heat recovery systems for FAD operations will produce significant energy and cost savings to the Army.

3. For waste heat recovery installations designed to preclude NG condensation, an automatic exhaust gas temperature control system is a desirable, if not mandatory, operational and safety feature. When properly adjusted, it maximizes energy recovery while minimizing the safety risk. Placing a lower limit on the exhaust side outlet temperature to avoid nitroglycerin condensation decreases the energy recovery effectiveness from about 62% (the full flow value) to about 40% for operation during the coldest months, when intake air flow is frequently bypassed.

4. If an appropriate NG collection device is installed (e.g., the system is used in conjunction with a solvent recovery scheme, such as that contemplated for modernized dry house facilities) nitroglycerin condensation can be permitted in the heat exchanger. This will increase the overall energy recovery effectiveness to the 62% value while also increasing the return on investment by about 10%.

RECOMMENDATIONS

The following system design modifications are recommended as a result of this demonstration:

1. There should be an automatic override of the alarm subsystem during startup and changeover procedures.

2. Leak-tight construction should be improved, especially door seals, consistent with safety considerations.

3. The exhaust temperature limit should be increased to at least 85°F (29.4°C) to preclude NG condensation.

4. A permanent separation should be installed between the primary steam coil drain and the intake air plenum to avoid ingesting air entrained moisture. An alternate approach would be to extend the drain pipe well beyond the air plenum to the vicinity of the walkway.

5. An aluminum or plastic deflector shield should be installed below the heat exchanger exhaust face to direct any NG condensate into the drip pan.

The heat pipe waste heat recovery technology developed under this contract for propellant forced-air dry houses should be incorporated into DARCOM Project 57X4462, Modernized FAD for Multi-Base Propellants.

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Table 1 FAD Waste Heat Recovery System Control Settings

	Dry cycle	Purge	Standby
Dry house air/steam controls			
1. Pneumatic switch (SW1)	DC	DC	SB
2. Primary coil steam valves			
(a) Dry cycle/purge (V1)	O	O	C
(b) Standby (V2)	C	C	O
Fan house			
1. Booster coil controller temp, °F (°C)	140 (60)	158 (70)	–
2. Mode select switch (SW2)	DC	P	SB

Legend

O - Open
C - Closed
DC - Dry cycle
P - Purge
SB - Standby

2717-001W

Table 2 NG vapor concentrations in typical propellant dry house exhausts

	ARRADCOM	RAAP Normal air flow	RAAP High air flow
NG vapor concentration, ppm max	2.5	2.6	18
Dry house temperature, ^{o}F (^{o}C)	140 (60)	140 (60)	140 (60)
Air flow rate through house, cfm (m^3/hr)	3000 (5100)	5500 (9350)	7300 (12400)
Type propellant dried	M7	M30	M30
Weight of propellant dried, lb (kg)	2500 (1134)	7344 (3331)	7344 (3331)
NG dew point (from figure 12), ^{o}F (^{o}C)	75 (23.9)	75 (23.9)	115 (46.1)

2717-002W

Table 3 Preparation and use of a typical NG killer (Ref 6)

Prepare approximately 20% sodium sulfide (Na_2S)/water solution. This solution will exothermically decompose NG.

Add the 20% sodium sulfide to an NG solvent (i.e., methanol or acetone) at an approximate weight ratio of 1:2 sulfide solution to solvent. The NG solvent/sulfide solution mixture should be completely miscible. The purpose of this solvent addition is to render the killer miscible with NG.

Precautions for the use of NG killer

1. The reaction between sodium sulfide and NG is exothermic and contact of the killer solution with appreciable quantities of NG will result in an explosion. Consequently, killer must be employed only after the area to be decontaminated has first been thoroughly washed down with an NG solvent or with detergent and water.
2. Killer solution fumes are extremely flammable and sparks or flames near any of the ingredients or the mixed killer should be avoided. All containers for handling the alcohol, acetone, and mixed killer should be grounded.
3. Procedures should require that preparation and application of killer be done only by Safety Department personnel.
4. Sulfide will precipitate from the killer solution if the solution is stored for any length of time. Once precipitation has occurred, the effectiveness of the killer is unknown. Consequently, fresh killer solution should be prepared immediately prior to its application. Unused solution should be discarded and not stored for future usage.

2717-003W

Table 4 FAD waste heat recovery test timeline

Test sequence	Start date/time	End date/time	Duration (hours)
Standby	1-27-78/1430	1-30-78/0930	67
Dry cycle No. 1	2-28-78/0925	3-1-78/1220	11 (warmup) 40 (dry cycle)
Dry cycle No. 2	3-7-78/1000	3-9-78/1300	11 (warmup) 40 (dry cycle)
Purge	3-13-78/1540	3-15-78/1540	48

2717-004W

Table 5 Data measurement list

(Refer to Figure 15 for schematic)

Symbol	Measurement
T1	HPHX intake air, inlet temperature
T2	HPHX intake air, outlet temperature
T3	Dry house air temperature
T4	Mixed intake air temperature
T5	Intake air after 300 lb primary steam coil
T6	Intake air before dry house booster coil
T7	Intake air after dry house booster coil
T8	HPHX exhaust air, inlet temperature
T9	HPHX exhaust air, outlet temperature
T10	Ambient air
T11	Old fan house duct, stagnant air
V_c	Duct centerline velocity (=0.9957 average velocity)
ΔP_D	Differential static pressure across HPHX fan
W-H	Kw-hour meter, HPHX fan

2717-005W

Table 6 Test equipment summary

Test equipment nomenclature	Manufacturer/model	Measurement comments
1. Thermocouples (11)	Thermoelectric TT-18-G-304-SST 1/8	Copper constantan T/C enclosed in grounded stainless steel sheath. Used to determine air temperature before and after each energy transfer device.
2. Air velocity system (1)	Sierra Instruments 435L-4-WT	Calibrated to indicate average inlet duct air velocity. Linear full scale output of 2 volts dc corresponds to full scale velocity of 2000 fpm. Reading times 1000 gives actual fpm.
3. Fan static pressure ΔP		Differential static pressure across fan, along with fan performance curves, used to evaluate fan operation. Linear full scale output of 2 volts dc corresponds to 4.0 in. water. Reading times 2.0 gives actual ΔP, in. water.
(a) Static pressure tubes	Dwyer, A-302	
(b) ΔP transducer	Robinson-Halpern 167A-W040	
4. Kilowatt-hour meter	Sangamo, S-25	Determines power consumption of HPHX Fan. Resolution 1 kw-hr.
5. Automatic data recorder	Kaye Instruments Multi-channel data logger	Permanent paper tape record of all transducer measurcments. Adjustable print interval. T/C data read directly, velocity and ΔP data are modified by calibration factors.

2717-006W

Table 7 Standby mode energy balance

Total energy requirement:	9.345 (10^6) Btu (2737.2 kw-hr)
Energy recovered by HPHX (balanced air flow)	
– intake side	2.407 (10^6) Btu (705 kw-hr)
– exhaust side	4.949 (10^6) Btu (1449.6 kw-hr)
– average	3.678 (10^6) Btu (1077.3 kw-hr)
Energy from steam coils (T7-T4)	6.986 (10^6) Btu (2049.2 kw-hr)
Fan motor	114 kw-hr = 0.389 (10^6) Btu
Energy recovery effectiveness	
– average	39.4%
– intake	25.8%

2717-007W

Table 8 Dry cycle No. 1 energy balance

	10^6 Btu (kw-hr)	
	Warmup	**Dry cycle**
Total energy requirement	2.737 (801.7)	14.251 (4174.1)
Energy recovered by HPHX (balanced air flow)		
– intake side	0.737 (215.9)	5.554 (1626.8)
– exhaust side	0.928 (271.8)	6.532 (1913.2)
– average	0.833 (244)	6.043 (1770)
Energy from steam coils (T7-T4)	2.241 (656.4)	9.059 (2653.4)
Fan motor	0.096 (28)	0.102 (30)
Energy recovery effectiveness		
– average	30.4%	42.4%
– intake	26.9%	39.0%

2717-008W

Table 9 Dry cycle No. 2 energy balance

	10^6 Btu (kw-hr)	
	Warmup	**Dry cycle**
Total energy requirement	2.604 (762.7)	13.350 (3910.2)
Energy recovered by HPHX (balanced air flow)		
– intake side	0.905 (265.1)	5.457 (1598.4)
– exhaust side	0.929 (272.1)	6.420 (1880.4)
– average	0.917 (268.6)	5.939 (1739.5)
Energy from steam coils (T7-T4)	2.019 (591.4)	8.389 (2457.1)
Fan motor	0.078 (23)	0.109 (32)
Energy recovery effectiveness		
– average	35.2%	44.5%
– intake	34.8%	40.9%

2717-009W

Table 10 Purge mode energy balance

Total energy requirement	15.548 (10^6) Btu (4554 kw-hr)
Energy recovered by HPHX	
– intake side	2.898 (10^6) Btu (848.8 kw-hr)
– exhaust side	2.591 (10^6) Btu (758.9 kw-hr)
– average	2.744 (10^6) Btu (803.7 kw-hr)
Energy from steam coils (T7-T4)	13.305 (10^6) Btu (3897 kw-hr)
Fan motor	100 kw-hr = 0.341 (10^6) Btu
Energy recovery effectiveness	
– average	17.6%
– intake	18.6%

2717-010W

Table 11 Summary of significant parameters for evaluation

Test Sequence	Recording Interval $\Delta\tau$ (min)	Duration (hours)	No. of data points n	ΣV	$\Sigma T3$	$\Sigma T10$	$\Sigma T4$	$\Sigma T7$	$\Sigma T9$	10^6 Btu	(kw-hr)
Standby	30	48	97	101778	7600	1703	3225	7624	4376	0.389	(114)
	30	67	134	142938	–	–	–	–	–	0.539	(158)
			127	–	–	–	4598	–	–		
			135	–	10603	2304	–	10711	6106		
Dry cycle No. 1											
Warmup	20	11	33	35025	3653	934	1669	3906	2697	0.096	(28)
Dry cycle	20	40	121	126577	17023	2828	8361	17402	10394	0.102	(30)
Dry cycle No. 2											
Warmup	20	11	33	34145	3691	1007	1948	4025	2708	0.078	(23)
Dry cycle	20	40	121	122446	16904	3122	8756	17420	10149	0.109	(32)
Purge											
at 12 hrs	20	12	37	39162	5171	1044	1767	5427	4462	0.085	(25)
at 24 hrs	20	24	73	76325	10220	2227	3685	10555	8877	0.171	(50)
at 48 hrs	20	48	145	150536	20351	4805	7663	21050	17609	0.351	(100)

2717-011W

Table 12 Average values of significant test parameters

Test sequence	Duration (hrs)	$\overline{V}$		$(A_D\overline{V})$		T3		T4		T7		T9		T10	
		fpm	m/s	cfm	m^3/s	°F	°C	°F	°C	°F	°C	°F	°C	°F	°C
1. Standby	48	1049	5.33	3022	1.43	78.4	25.8	33.2	0.7	78.6	25.9	45.1	7.3	17.6	-8.0
2. Dry cycle No. 1															
– Warmup	11	1062	5.39	3057	1.44	110.7	43.7	50.6	10.3	118.4	48.0	81.7	27.6	28.3	-2.1
– Dry cycle	40	1046	5.31	3013	1.42	140.7	60.4	69.1	20.6	143.8	62.1	85.9	29.9	23.4	-4.8
3. Dry cycle No. 2															
– Warmup	11	1035	5.26	2980	1.41	111.8	44.3	59.0	15.0	122.0	50.0	82.1	27.8	30.5	-0.8
– Dry cycle	40	1012	5.14	2915	1.38	139.7	59.8	72.4	22.4	144.0	62.2	83.9	28.8	25.8	-3.4
4. Purge															
at 12 hrs	12	1058	5.37	3048	1.44	139.8	59.9	47.8	8.8	146.7	63.7	120.6	49.2	28.2	-2.1
at 24 hrs	24	1046	5.31	3011	1.42	140.0	60.0	50.5	10.3	144.6	62.6	121.6	49.8	30.5	-0.8
at 48 hrs	48	1038	5.27	2990	1.41	140.4	60.2	52.8	11.6	145.2	62.9	121.4	49.7	33.1	0.6

Table 13 Overall system energy balance summary

Test Sequence	Air Mass Flow Rate		Total Energy Reqm't 10^6 Btu (kw-hr)	Energy Recovered by HPHX 10^6 Btu (kw-hr)		Energy From Steam Coils, 10^6 Btu (kw-hr)	Fan motor		Energy recovery effectiveness	
	lb/hr	$\frac{kg}{hr}$		intake	Exhaust ($m_x = m_t$)		10^6 Btu	(kw-hr)	intake	average
1. Standby (48 hr)	13,345	6059	9.345 (2737.2)	2.407 (705)	4.949 (1449.6)	6.986 (2046.2)	0.491	(114)	25.8	39.4
2. Dry cycle No. 1										
a) Warmup (11 hr)	13,023	5912	2.737 (801.7)	.737 (215.9)	.928 (271.8)	2.241 (656.4)	0.096	(28)	26.1	30.4
b) Dry cycle (40 hr)	12,655	5745	14.251 (4174.1)	5.554 (1626.8)	6.532 (1913.2)	9.059 (2653.4)	0.102	(30)	39.0	42.0
3. Dry cycle No. 2										
a) Warmup (11 hr)	12,480	5666	2.604 (762.7)	.905 (265.1)	.929 (272.1)	2.019 (591.4)	0.078	(23)	34.8	35.2
b) Dry cycle (40 hr)	12,208	5542	13.350 (3910.2)	5.457 (1598.4)	6.420 (1880.4)	8.389 (2457.1)	0.109	(32)	40.9	44.3
4. Purge (48 hr)	12,558	5701	15.548 (4554)	2.898 (848.8)	2.591 (758.9)	13.305 (3897)	0.341	(100)	18.6	17.6

2717-013W

Table 14 Nitroglycerin sensitivity test results

Operating mode	HX exhaust surfaces (fins)	Fan plenum: Access doors	Fan plenum: Intake screen	Air filters
	Location			
Dry cycle No. 1	Pos	Neg	Neg	Neg
Dry cycle No. 2	Pos	Pos	Neg	Pos
Purge, after				
– 18 hr	Pos	–	–	–
– 21 hr	Pos	–	–	–
– 24 hr	Weak pos	–	–	–
– 48 hr	Neg	–	–	–

2717-014W

Table 15 Economic analysis, existing dry house, no NG condensation, one system per bay

Project/Program Costs (Millions $)							
1. Project year (FY)	2. Non-recurring costs (alternative)	3. Recurring/operating costs (status quo)	4. Recurring/operating costs (alternative)	5. Differential annual costs (annual cost savings) Cols 3. Less 4.	6. Discount factor	7. Non-recurring costs discounted Cols 2. x 6.	8. Differential annual costs discounted Cols 5. x 6.
79	2.789				0.954	2.661	
80		1.909	0.904	1.005	0.867		0.871
81		2.004	0.949	1.055	0.788		0.831
82		2.104	0.997	1.107	0.717		0.794
83		2.210	1.046	1.164	0.652		0.759
84		2.320	1.099	1.221	0.592		0.723
85		2.436	1.154	1.282	0.538		0.690
86		2.558	1.211	1.347	0.489		0.659
87		2.686	1.272	1.414	0.445		0.629
88		2.820	1.336	1.484	0.405		0.601
89		2.961	1.402	1.559	0.368		0.574
					0.334		
					0.304		
					0.276		
					0.251		
Totals						2.661	7.131

9. Col 7. Total cost	$2.661
10. Adjustments to Col 7. total	
(a) Add: Discounted value of existing assets to be used on the proposed project	(+) 0
(b) Subtract: Discounted value of existing assets replaced by the proposed project	(-) 0
(c) Subtract: Discounted terminal value of the new investment	(-) 0
11. Total adjusted discounted non-recurring costs (sum of 9 & 10. (a), less 10. (b) & 10. (c)	2.661
12. Col 8. Total cost	$7.131
13. Adjustment to Col 8. total Add: Discounted value of expansion, modification or overhaul eliminated by the new investment	(+) 0
14. Total adjusted annual differential cost (sum of 12. & 13. (a))	7.131
15. Savings/investment ratio (Item 14. ÷ by Item 11.)	2.680
16. Return on investment (ROI)	38.8 %

2717-015W

Table 16 Economic analysis, existing dry house, with NG condensation, one system per bay

Project/Program Costs (Millions $)							
1. Project year (FY)	2. Non-recurring costs (alternative)	3. Recurring/operating costs (status quo)	4. Recurring/operating costs (alternative)	5. Differential annual costs (annual cost savings) Cols 3. Less 4.	6. Discount factor	7. Non-recurring costs discounted Cols 2. x 6.	8. Differential annual costs discounted Cols 5. x 6.
79	2.328				0.954	2.221	
80		1.909	0.806	1.103	0.867		0.956
81		2.004	0.847	1.157	0.788		0.912
82		2.104	0.889	1.215	0.717		0.871
83		2.210	0.933	1.277	0.652		0.833
84		2.320	0.980	1.340	0.592		0.793
85		2.436	1.029	1.407	0.538		0.750
86		2.558	1.080	1.478	0.489		0.723
87		2.686	1.134	1.552	0.445		0.691
88		2.820	1.191	1.629	0.405		0.660
89		2.961	1.251	1.710	0.368		0.629
					0.334		
					0.304		
					0.276		
					0.251		
Totals						2.221	7.818

2717-016W

9. Col 7. Total cost — $2.221

10. Adjustments to Col 7. total
    (a) Add: Discounted value of existing assets to be used on the proposed project — (+) 0
    (b) Subtract: Discounted value of existing assets replaced by the proposed project — (-) 0
    (c) Subtract: Discounted terminal value of the new investment — (-) 0

11. Total adjusted discounted non-recurring costs (sum of 9 & 10. (a), less 10. (b) & 10. (c) — 2.221

12. Col 8. Total cost — $7.818

13. Adjustment to Col 8. total Add: Discounted value of expansion, modification or overhaul eliminated by the new investment — (+) 0

14. Total adjusted annual differential cost (sum of 12. & 13. (a)) — 7.818

15. Savings/investment ratio (Item 14. ÷ by Item 11.) — 3.520

16. Return on investment (ROI) — 51.1%

Table 17 Economic analysis, existing dry house, no NG condensation, one system per two bays

Project/Program Costs (Millions $)							
1. Project year (FY)	2. Non-recurring costs (alternative)	3. Recurring/operating costs (status quo)	4. Recurring/operating costs (alternative)	5. Differential annual costs (annual cost savings) Cols 3. Less 4.	6. Discount factor	7. Non-recurring costs discounted Cols 2. x 6.	8. Differential annual costs discounted Cols 5. x 6.
79	1.930				0.954	1.841	
80		1.909	0.904	1.005	0.867		0.871
81		2.004	0.949	1.055	0.788		0.831
82		2.104	0.997	1.107	0.717		0.794
83		2.210	1.046	1.164	0.652		0.759
84		2.320	1.099	1.221	0.592		0.723
85		2.436	1.154	1.282	0.538		0.690
86		2.558	1.211	1.347	0.489		0.659
87		2.686	1.272	1.414	0.445		0.629
88		2.820	1.336	1.484	0.405		0.601
89		2.961	1.402	1.559	0.368		0.574
					0.334		
					0.304		
					0.276		
					0.251		
Totals						1.841	7.131

2717-017W

9. Col 7. Total cost — $1.841

10. Adjustments to Col 7. total
 (a) Add: Discounted value of existing assets to be used on the proposed project — (+) 0
 (b) Subtract: Discounted value of existing assets replaced by the proposed project — (-) 0
 (c) Subtract: Discounted terminal value of the new investment — (-) 0

11. Total adjusted discounted non-recurring costs (sum of 9 & 10. (a), less 10. (b) & 10. (c) — 1.841

12. Col 8. Total cost — $7.131

13. Adjustment to Col 8. total Add: Discounted value of expansion, modification or overhaul eliminated by the new investment — (+) 0

14. Total adjusted annual differential cost (sum of 12. & 13. (a)) — 7.131

15. Savings/investment ratio (Item 14. ÷ by Item 11.) — 3.873

16. Return on investment (ROI) — 56.1 %

Table 18 Economic analysis, existing dry house, with NG condensation, one system per two bays

Project/Program Costs (Millions $)							
1. Project year (FY)	2. Non-recurring costs (alternative)	3. Recurring/operating costs (status quo)	4. Recurring/operating costs (alternative)	5. Differential annual costs (annual cost savings) Cols 3. Less 4.	6. Discount factor	7. Non-recurring costs discounted Cols 2. x 6.	8. Differential annual costs discounted Cols 5. x 6.
79	1.699				0.954	1.621	
80		1.909	0.806	1.103	0.867		0.956
81		2.004	0.847	1.157	0.788		0.912
82		2.104	0.889	1.215	0.717		0.871
83		2.210	0.933	1.277	0.652		0.833
84		2.320	0.980	1.340	0.592		0.793
85		2.436	1.029	1.407	0.538		0.750
86		2.558	1.080	1.478	0.489		0.723
87		2.686	1.134	1.552	0.445		0.691
88		2.820	1.191	1.629	0.405		0.660
89		2.961	1.251	1.710	0.368		0.629
					0.334		
					0.304		
					0.276		
					0.251		
Totals						1.621	7.818

2717-018W

9. Col. 7. Total cost — $1.621
10. Adjustments to Col 7. total
    (a) Add: Discounted value of existing assets to be used on the proposed project — (+) 0
    (b) Subtract: Discounted value of existing assets replaced by the proposed project — (-) 0
    (c) Subtract: Discounted terminal value of the new investment — (-) 0
11. Total adjusted discounted non-recurring costs (sum of 9 & 10. (a), less 10. (b) & 10. (c) — 1.621
12. Col 8. Total cost — $7.818
13. Adjustment to Col 8. total Add: Discounted value of expansion, modification or overhaul eliminated by the new investment — (+) 0
14. Total adjusted annual differential cost (sum of 12. & 13. (a)) — 7.818
15. Savings/investment ratio (Item 14. ÷ by Item 11.) — 4.823
16. Return on investment (ROI) — 69.3 %

Table 19 Economic analysis, existing dry house, with NG condensation, one system per bay

Project/Program Costs (Millions $)							
1. Project year (FY)	2. Non-recurring costs (alternative)	3. Recurring/operating costs (status quo)	4. Recurring/operating costs (alternative)	5. Differential annual costs (annual cost savings) Cols 3. Less 4.	6. Discount factor	7. Non-recurring costs discounted Cols 2. x 6.	8. Differential annual costs discounted Cols 5. x 6.
79	1.622				0.954	1.547	
80		0.694	0.293	0.401	0.867		0.348
81		0.729	0.308	0.421	0.788		0.332
82		0.765	0.323	0.442	0.717		0.317
83		0.804	0.339	0.465	0.652		0.303
84		0.844	0.356	0.488	0.592		0.289
85		0.886	0.374	0.512	0.538		0.275
86		0.930	0.393	0.537	0.489		0.263
87		0.977	0.413	0.564	0.445		0.251
88		1.026	0.433	0.593	0.405		0.240
89		1.077	0.455	0.622	0.368		0.229
					0.334		
					0.304		
					0.276		
					0.251		
Totals						1.547	2.847

2717-019W

9. Col. 7. Total cost — $1.547

10. Adjustments to Col 7. total
    (a) Add: Discounted value of existing assets to be used on the proposed project — (+) 0
    (b) Subtract: Discounted value of existing assets replaced by the proposed project — (-) 0
    (c) Subtract: Discounted terminal value of the new investment — (-) 0

11. Total adjusted discounted non-recurring costs (sum of 9 & 10. (a), less 10. (b) & 10. (c) — 1.547

12. Col 8. Total cost — $2.847

13. Adjustment to Col 8. total Add: Discounted value of expansion, modification or overhaul eliminated by the new investment — (+) 0

14. Total adjusted annual differential cost (sum of 12. & 13. (a)) — 2.847

15. Savings/investment ratio (Item 14. ÷ by Item 11.) — 1.840

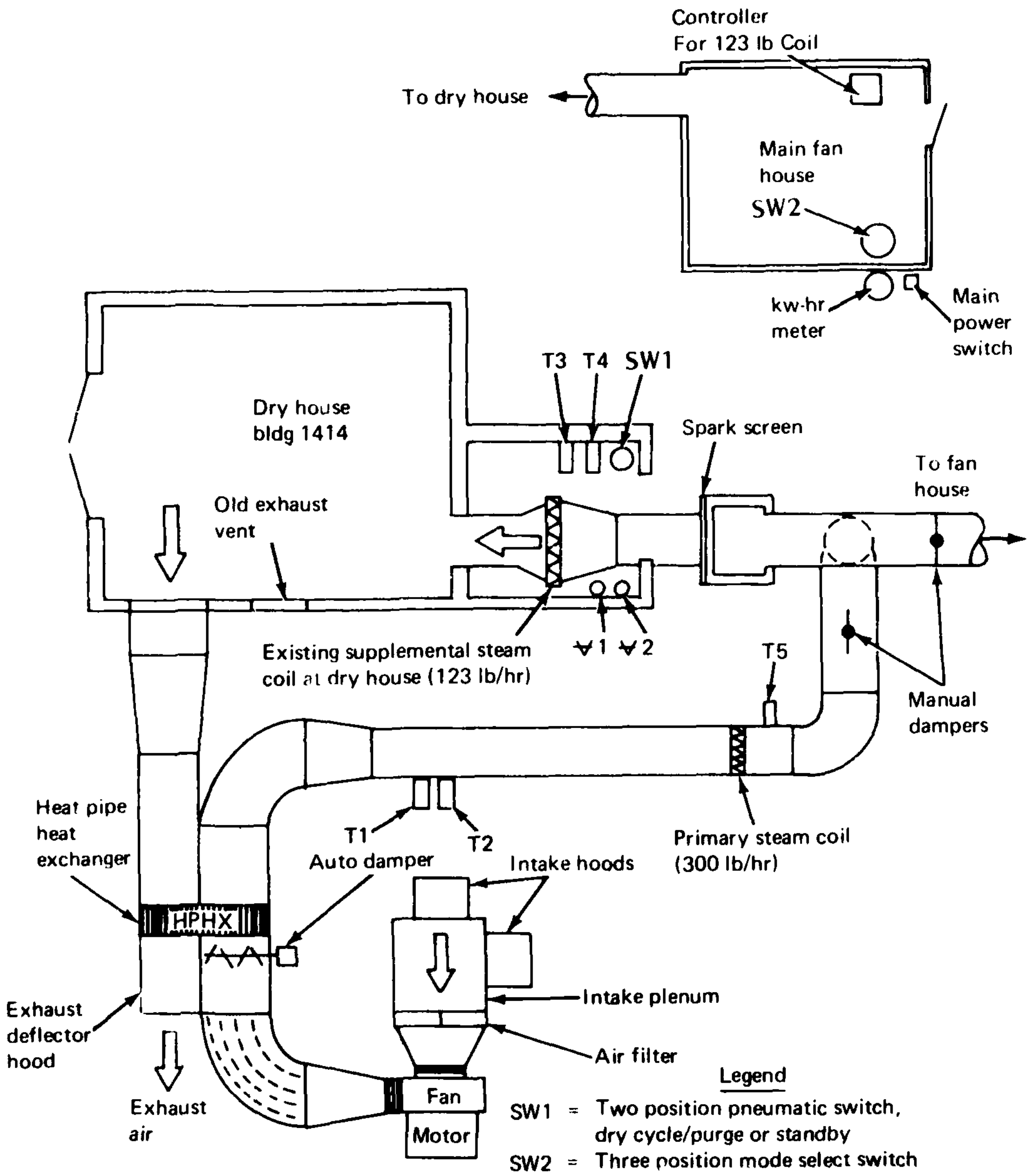
16. Return on investment (ROI) — 25.6 %

Table 20 Economic analysis, modernized dry house, with NG condensation, one system per two bays

Project/Program Costs (Millions $)							
1. Project year (FY)	2. Non-recurring costs (alternative)	3. Recurring/operating costs (status quo)	4. Recurring/operating costs (alternative)	5. Differential annual costs (annual cost savings) Cols 3. Less 4.	6. Discount factor	7. Non-recurring costs discounted Cols 2. x 6.	8. Differential annual costs discounted Cols 5. x 6.
79	0.955				0.954	0.911	
80		0.694	0.293	0.401	0.867		0.348
81		0.729	0.308	0.421	0.788		0.332
82		0.765	0.323	0.442	0.717		0.317
83		0.804	0.339	0.465	0.652		0.303
84		0.844	0.356	0.488	0.592		0.289
85		0.886	0.374	0.512	0.538		0.275
86		0.930	0.393	0.537	0.489		0.263
87		0.977	0.413	0.564	0.445		0.251
88		1.026	0.433	0.593	0.405		0.240
89		1.077	0.455	0.622	0.368		0.229
					0.334		
					0.304		
					0.276		
					0.251		
Totals						0.911	2.847

2717-020W

9.	Col 7. Total cost	$0.911
10.	Adjustments to Col 7 total	
	(a) Add: Discounted value of existing assets to be used on the proposed project	(+) 0
	(b) Subtract: Discounted value of existing assets replaced by the proposed project	(-) 0
	(c) Subtract: Discounted terminal value of the new investment	(-) 0
11.	Total adjusted discounted non-recurring costs (sum of 9 & 10. (a), less 10. (b) & 10. (c)	0.911
12	Col 8. Total cost	$2.847
13.	Adjustment to Col 8. total Add: Discounted value of expansion, modification or overhaul eliminated by the new investment	(+) 0
14	Total adjusted annual differential cost (sum of 12 & 13. (a))	2.847
15	Savings/investment ratio (Item 14 by Item 11.)	3.125
16	Return on investment (ROI)	45.4 %



Legend

SW1	=	Two position pneumatic switch, dry cycle/purge or standby
SW2	=	Three position mode select switch for alarms
T1	=	Low temp alarm thermostat, standby
T2	=	Low temp alarm thermostat, dry cycle/purge
T3	=	Auto damper thermostat, dry cycle/purge
T4	=	Auto damper thermostat, standby
T5	=	High temperature alarm
V1	=	Steam valve, dry cycle and purge
V2	=	Steam valve, standby

2717-021W

Fig. 1 Recovery system schematic

2717-022W

Fig. 2 Finished recovery system, including duct work

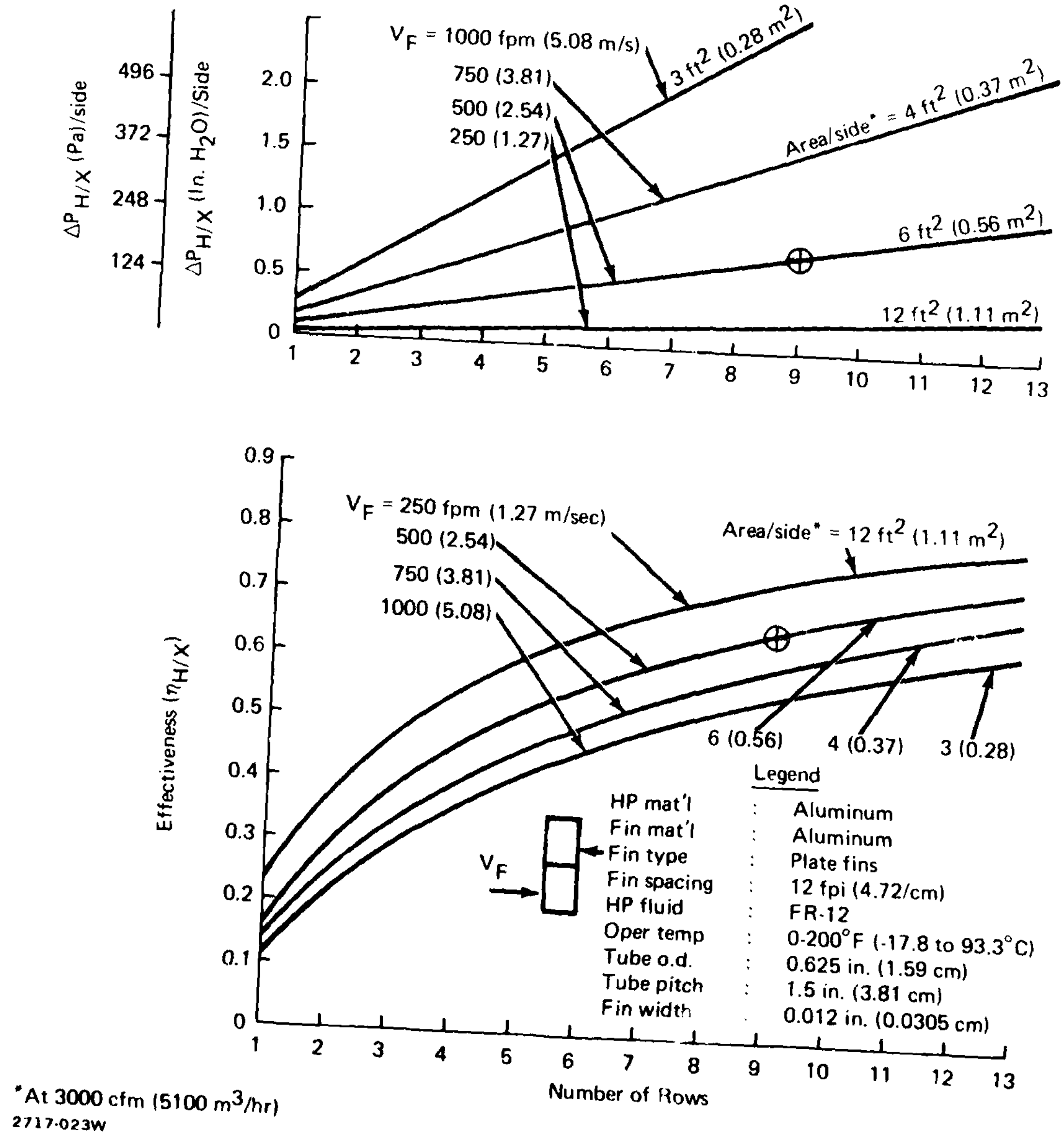
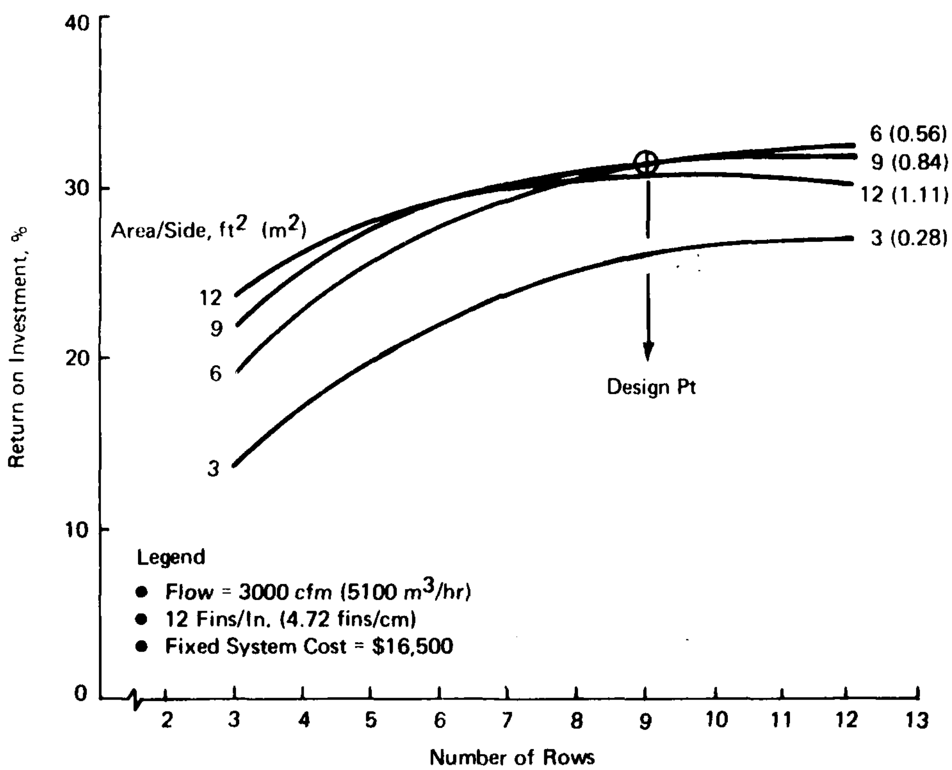
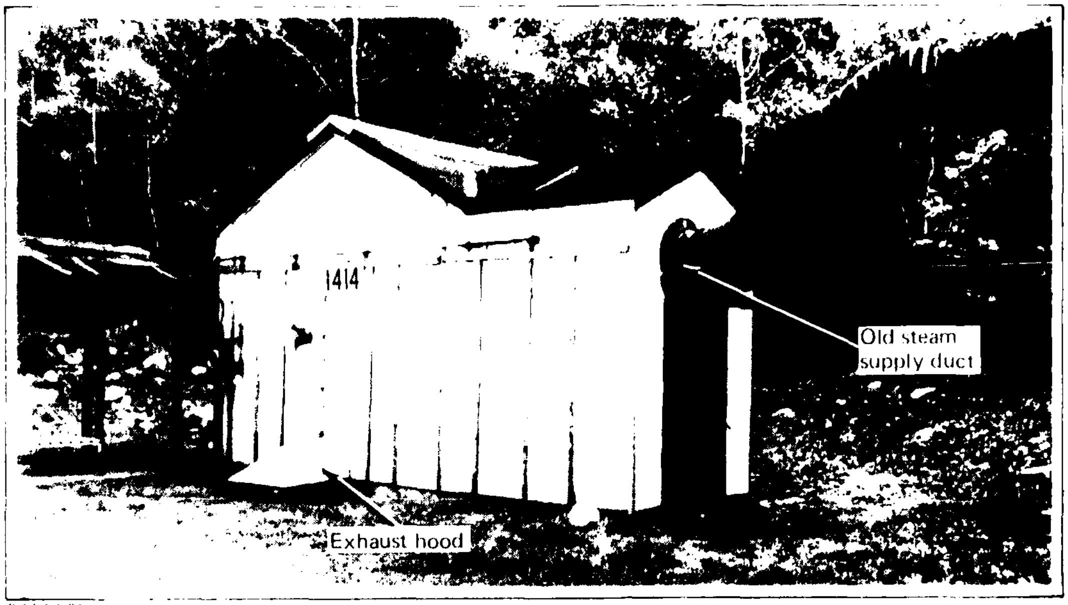


Fig. 3 Heat pipe heat exchanger performance



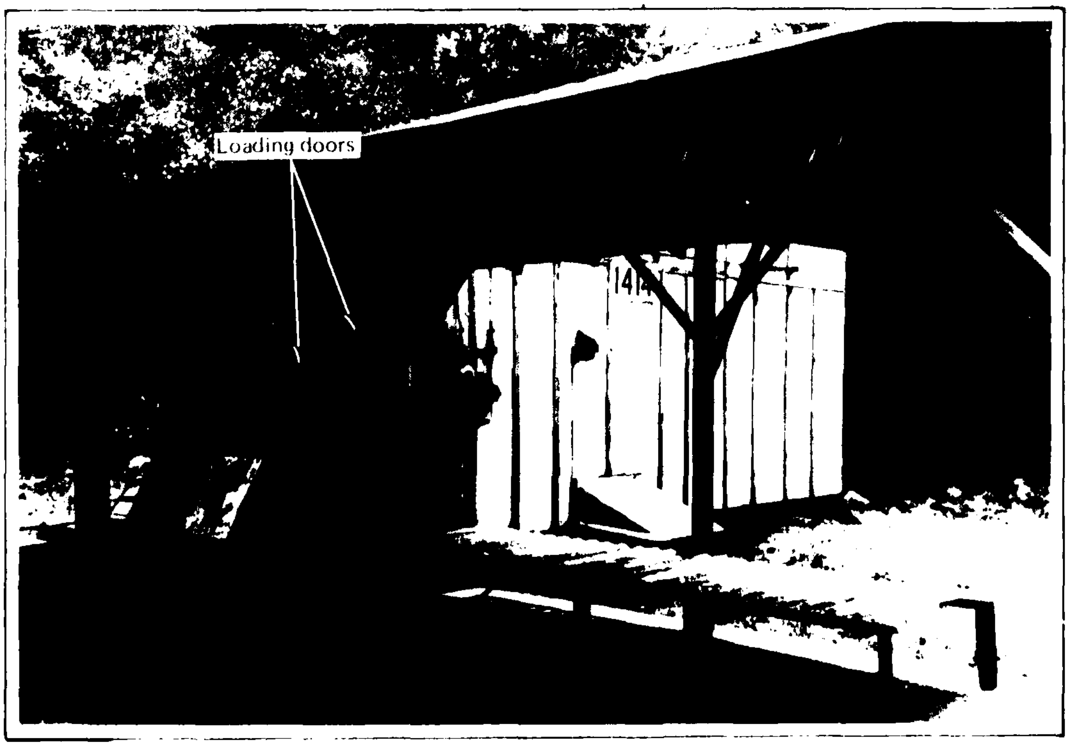
2717-024W

Fig. 4 Return on investment vs HPHX size



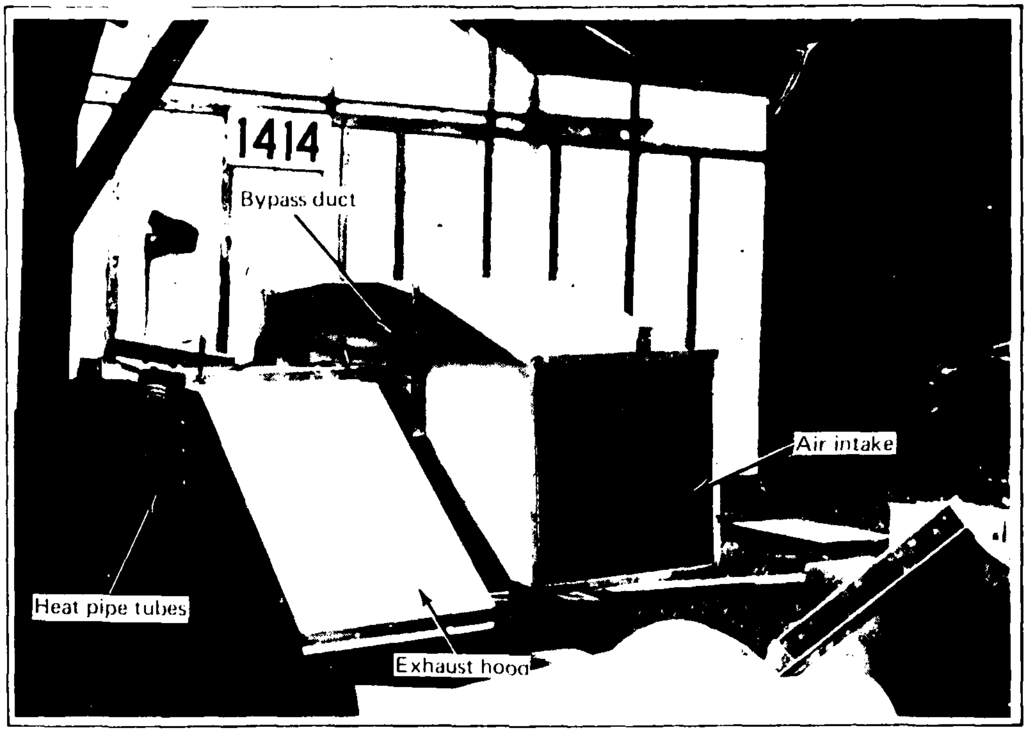
2717-025W

Fig. 5 Forced-air dry house before modification (right side view)



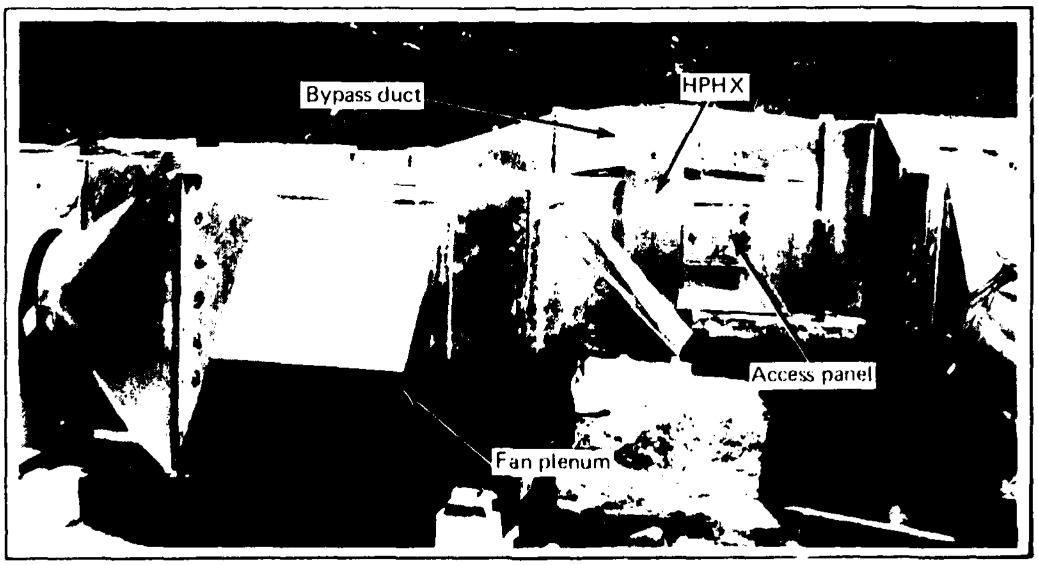
2717-026W

Fig. 6 Forced-air dry house before modification (left side view)



2717-027W

Fig. 7 Heat pipe heat exchanger and exhaust canopy



2717-028W

Fig. 8 Recovery unit before insulation was applied

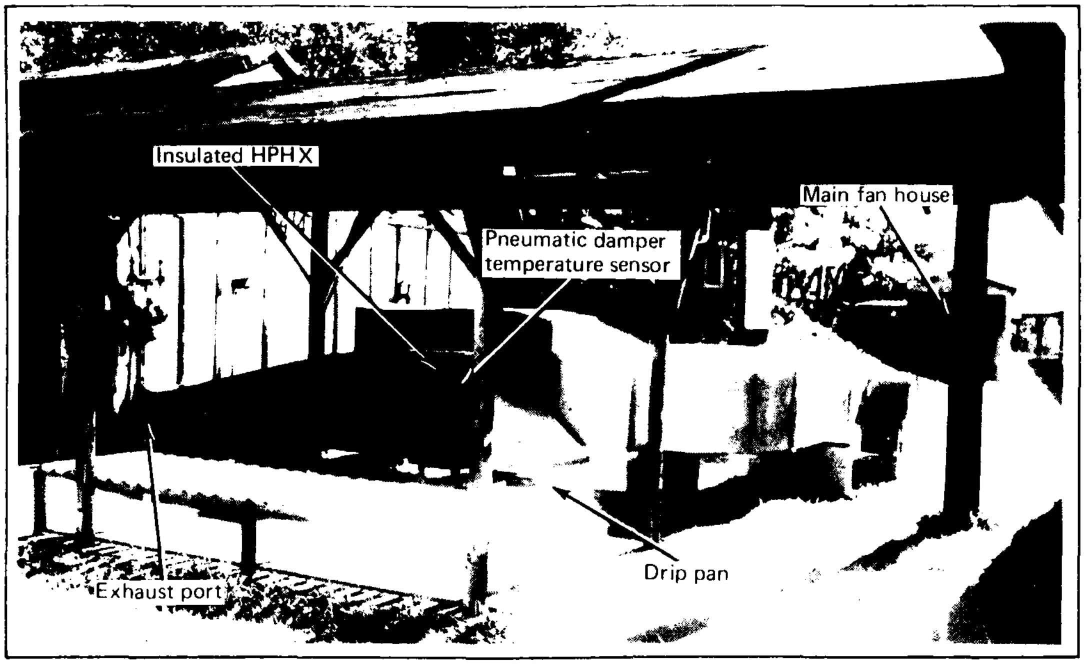


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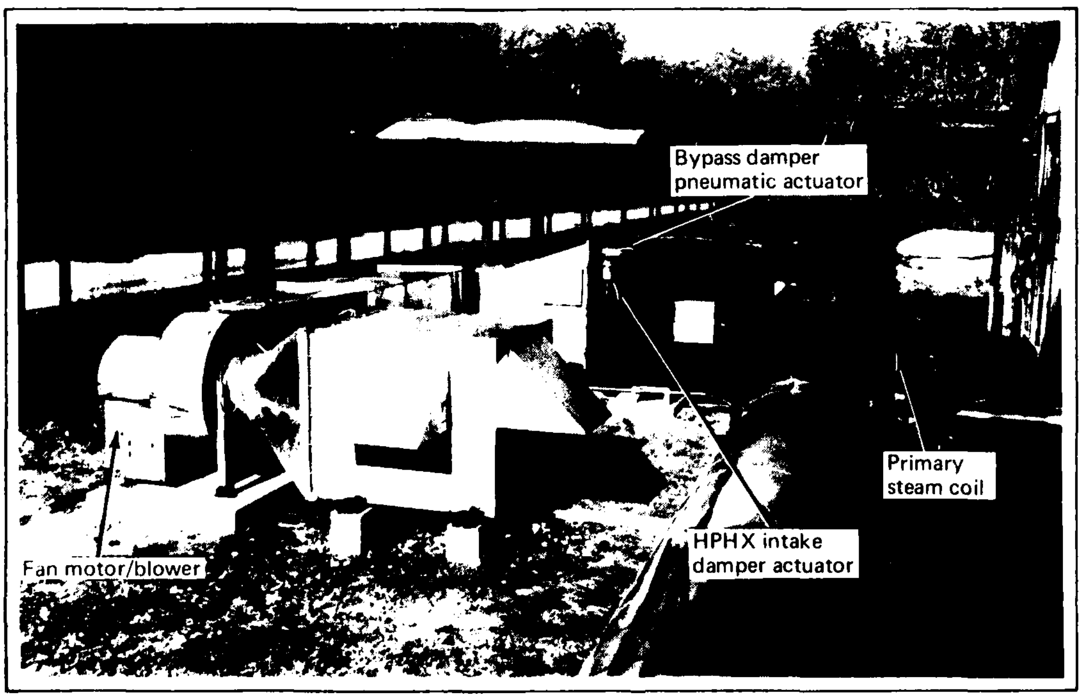
Fig. 9 Finished recovery unit (left side view)

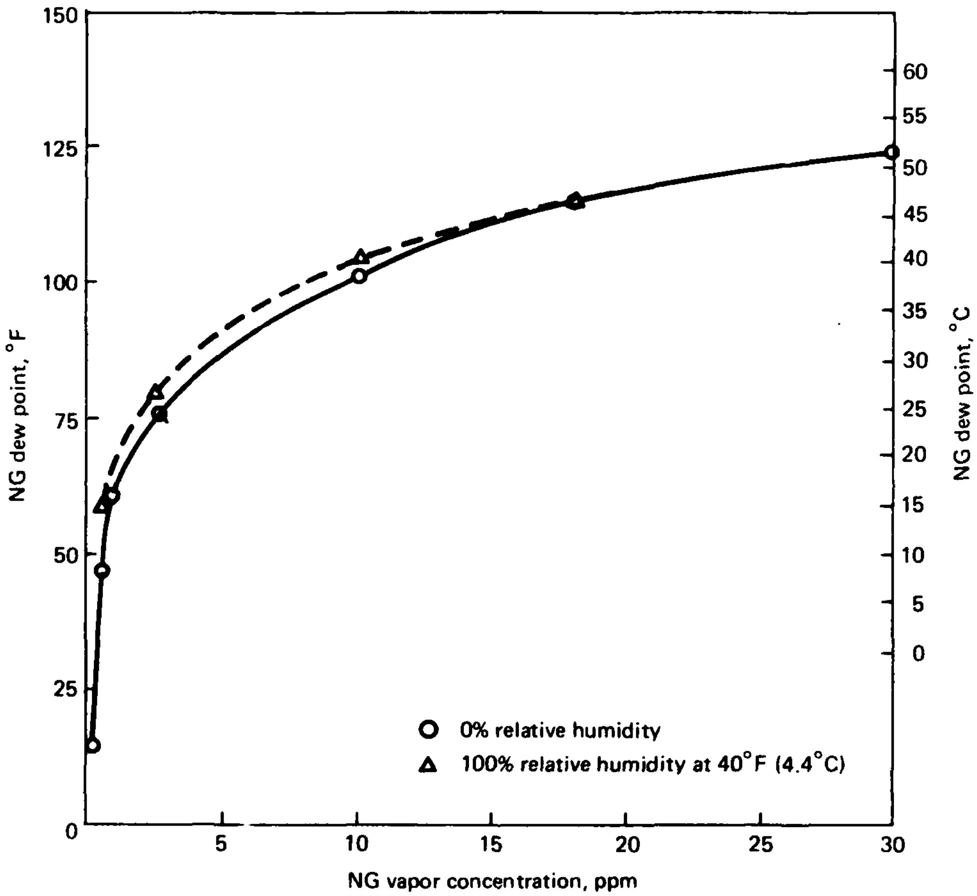


2717-030W

Fig. 10 Finished recovery unit showing steam trap for primary steam coil

2717-031W

Fig. 11 Finished recovery unit (right side view)



2717-032W

Fig. 12 Dew point of NG vapor-laden air vs vapor concentration based on vapor pressure data of Crater (Ref 3) and 30 in. (760 mm) Hg atmospheric pressure

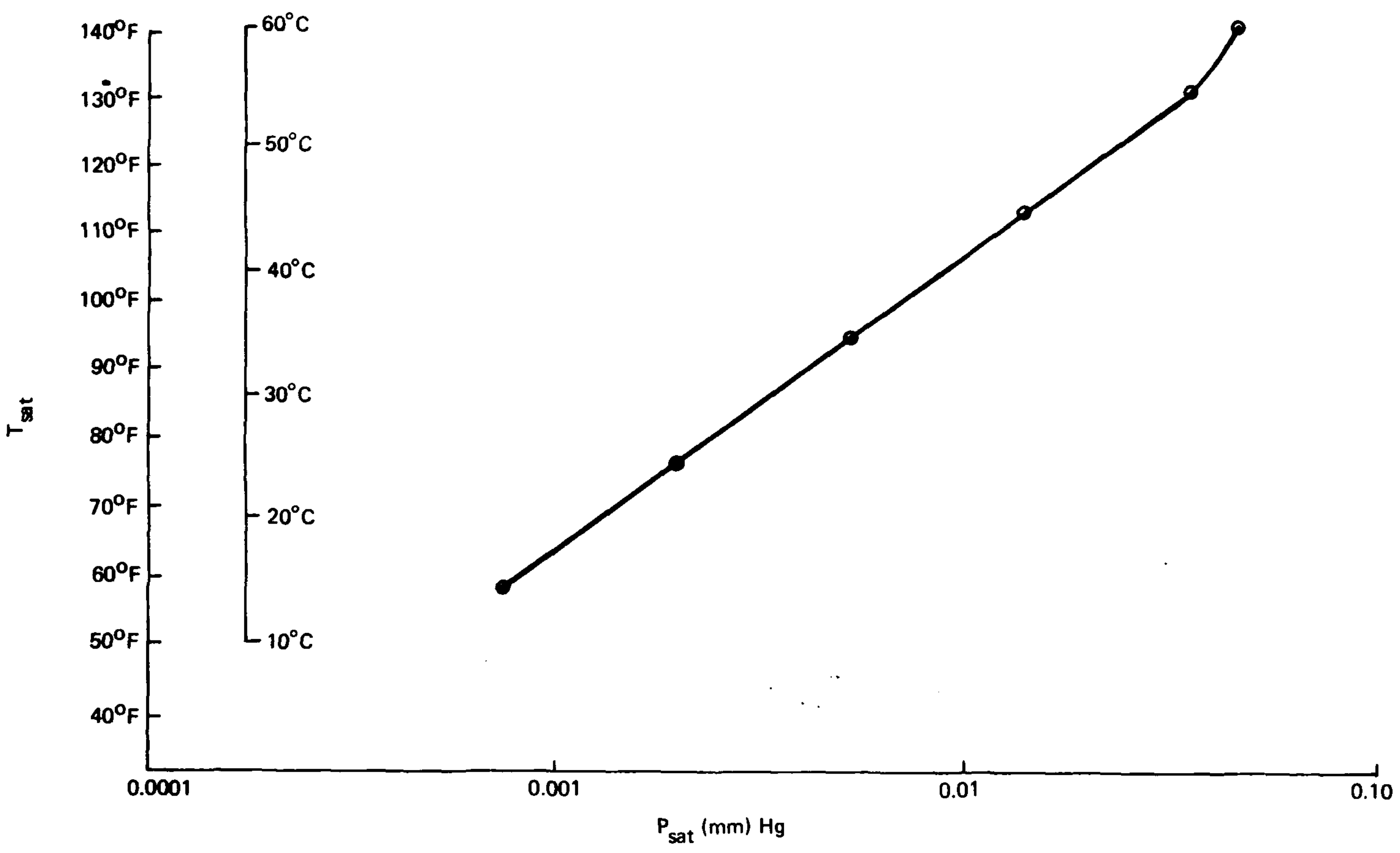


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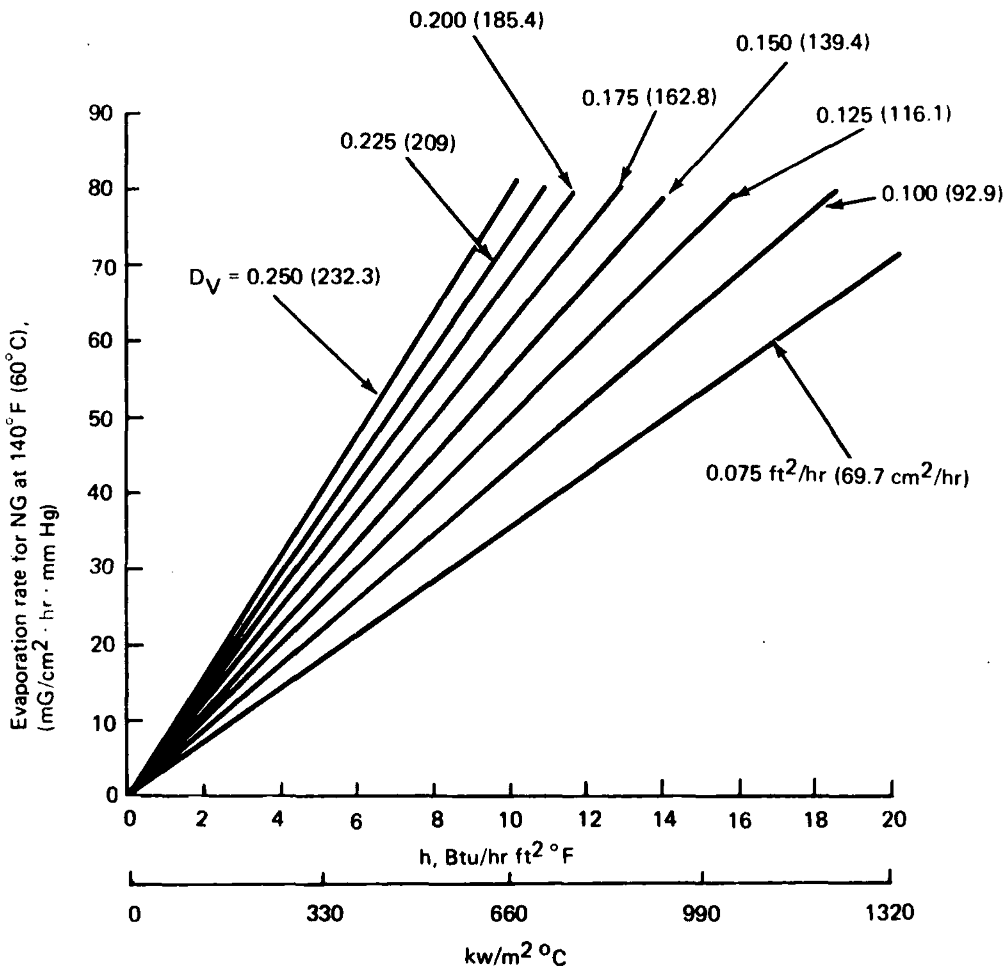
Fig. 13 Nitroglycerin saturation data, P vs T (Ref 3)



2717-034W

Fig. 14 Nitroglycerin evaporation rates at 140°F (60°C) as a function of bulk air side film heat transfer coefficient and diffusivity

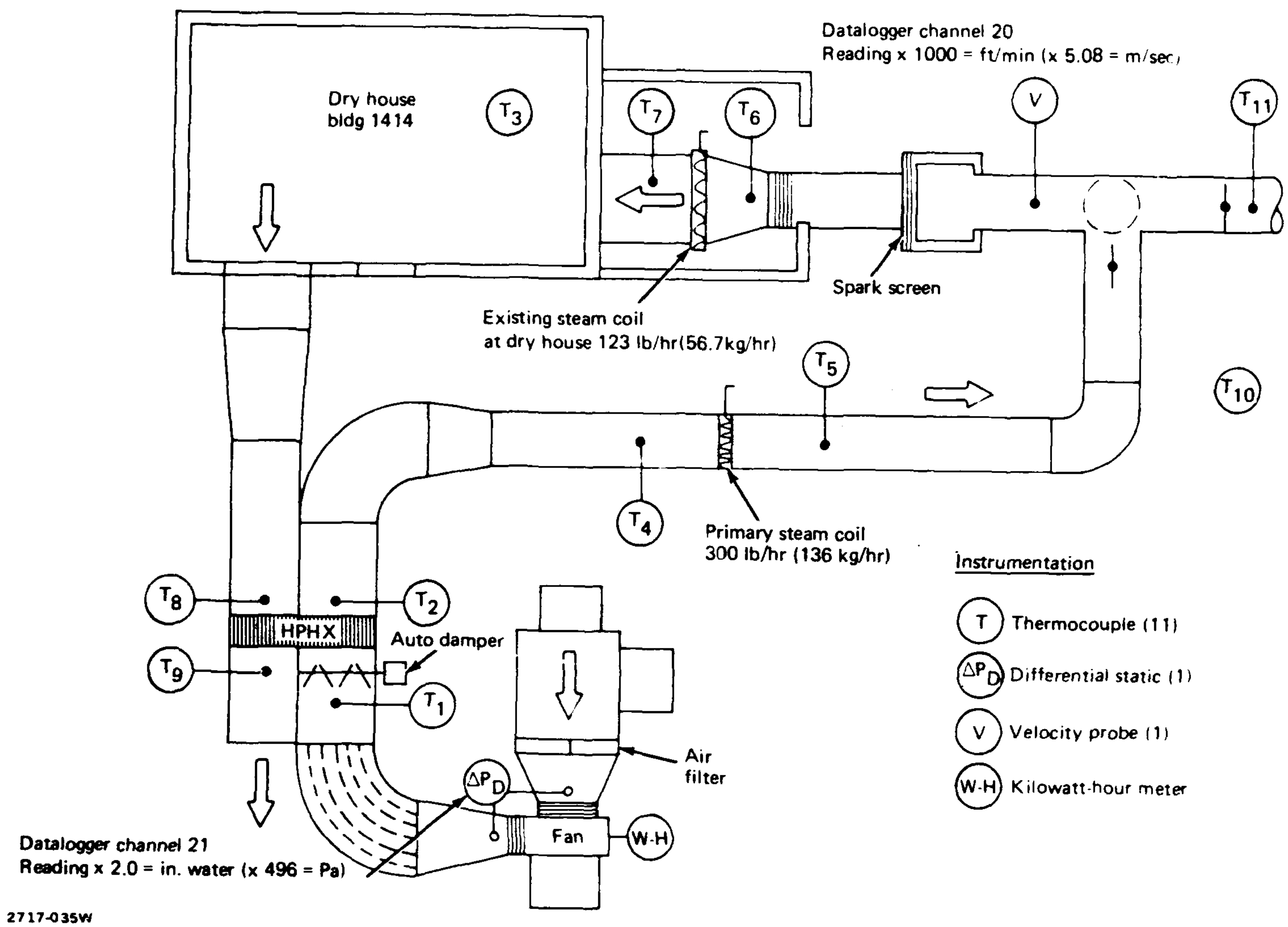


Fig. 15 Dry house waste heat recovery system instrumentation

	021	1.4974	V	← Fan ΔP (reading x 2.0 = in. of water)
	020	- 1.0058	V	← Average duct velocity (reading x 1000 = fpm)
Temperatures per figure 15	011	0067.8	F	
	010	0020.8	F	
	009	0047.4	F	
	008	0079.4	F	
	007	0065.9	F	
	006	0063.9	F	
	005	0049.4	F	
	004	0039.0	F	
	003	0079.2	F	
	002	0043.9	F	
	001	0021.3	F	
Time → 15 20 01	000	0045.3	F	

2717-036W

Fig. 16 Sample data record

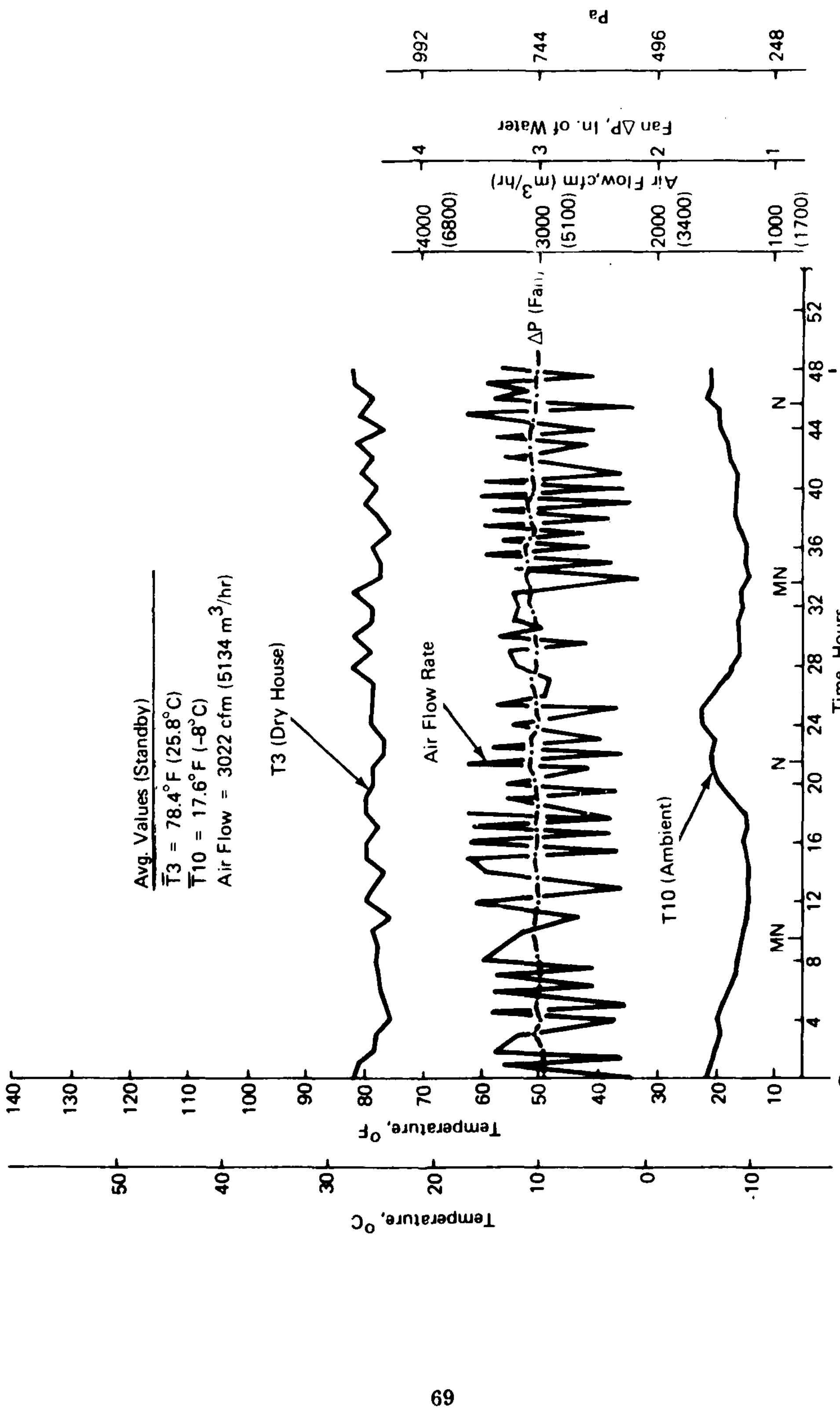
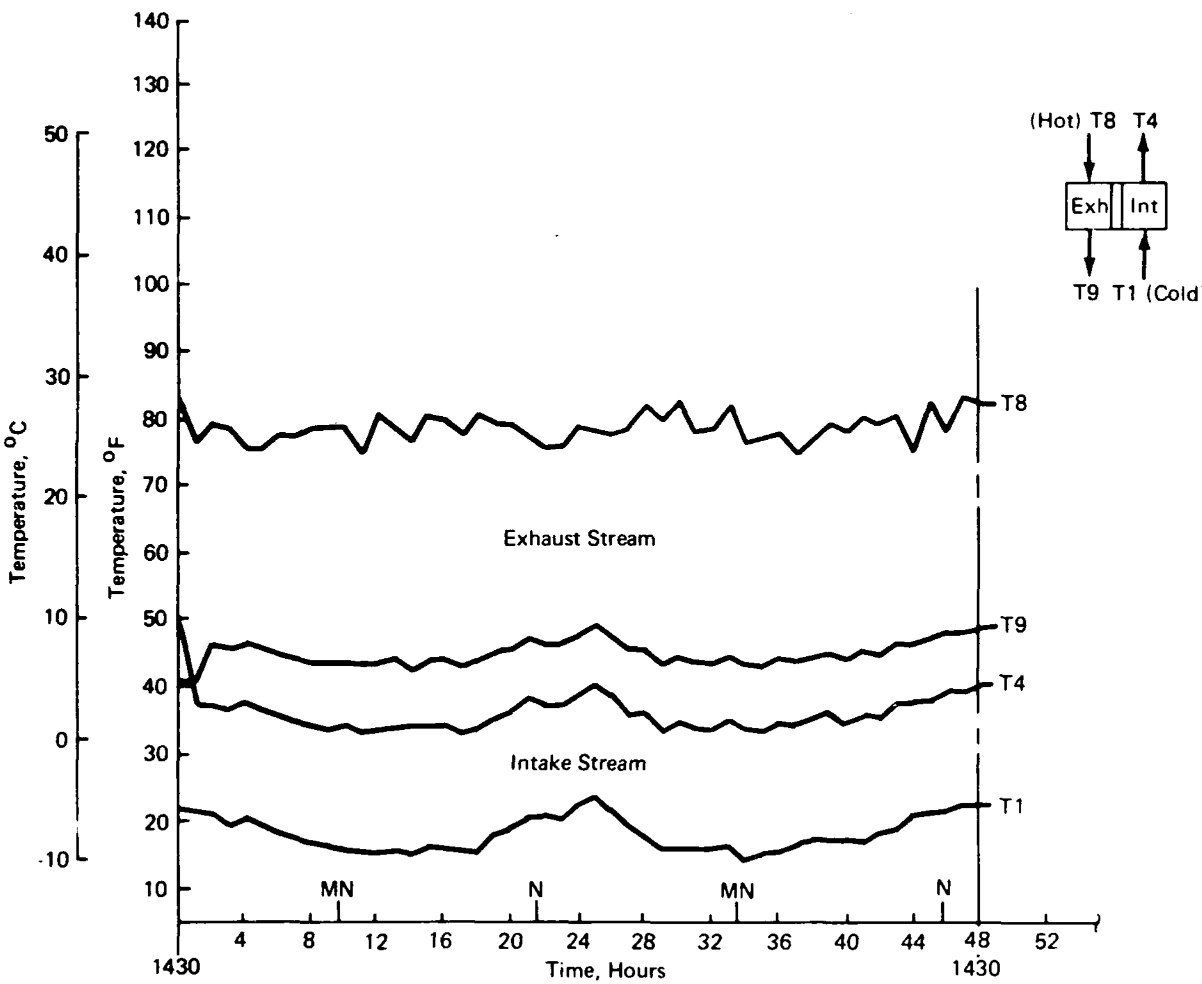


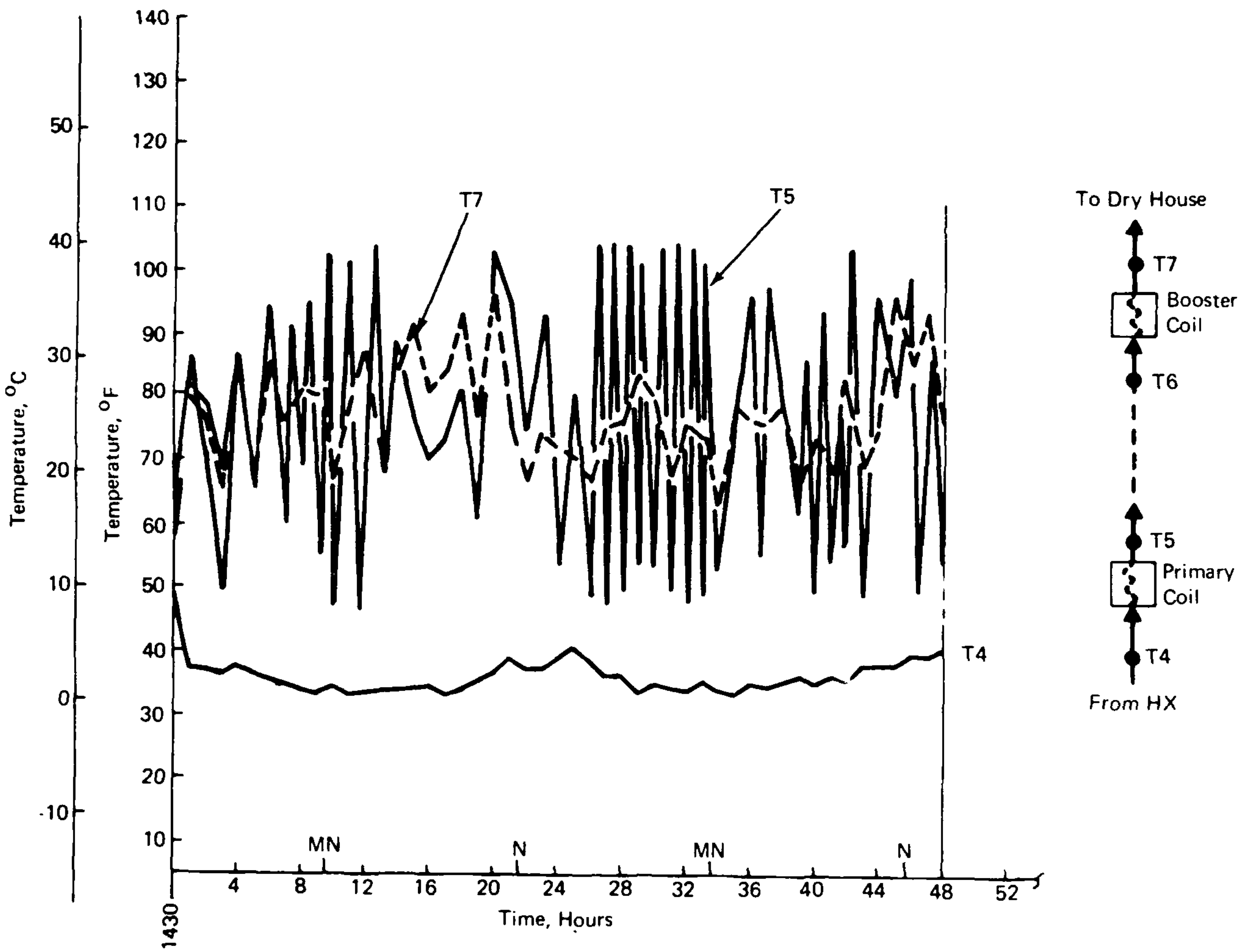
Fig. 17 Overall system performance (standby)

2717-037W



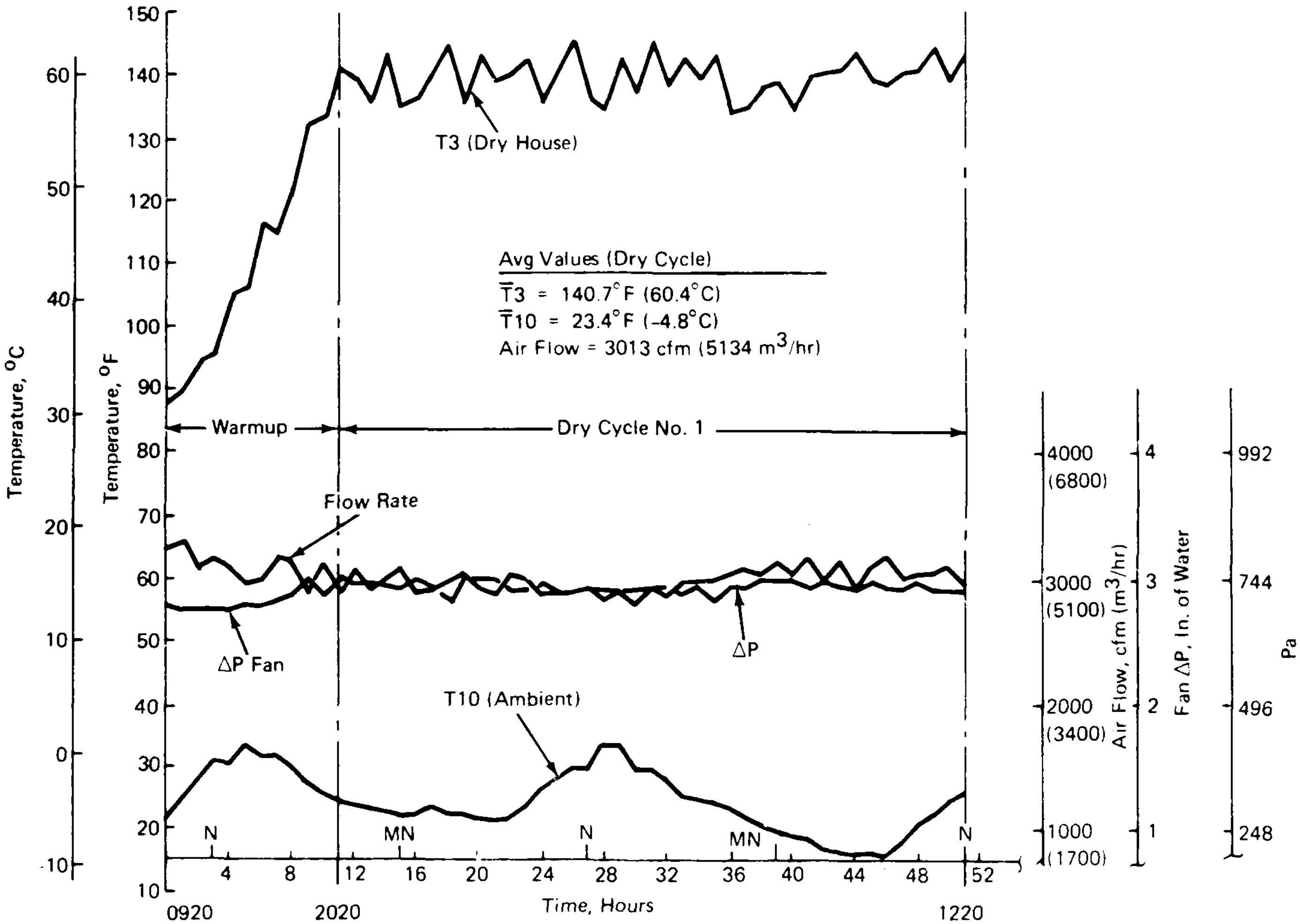
2717-038W

Fig. 18 Heat exchanger performance (standby)



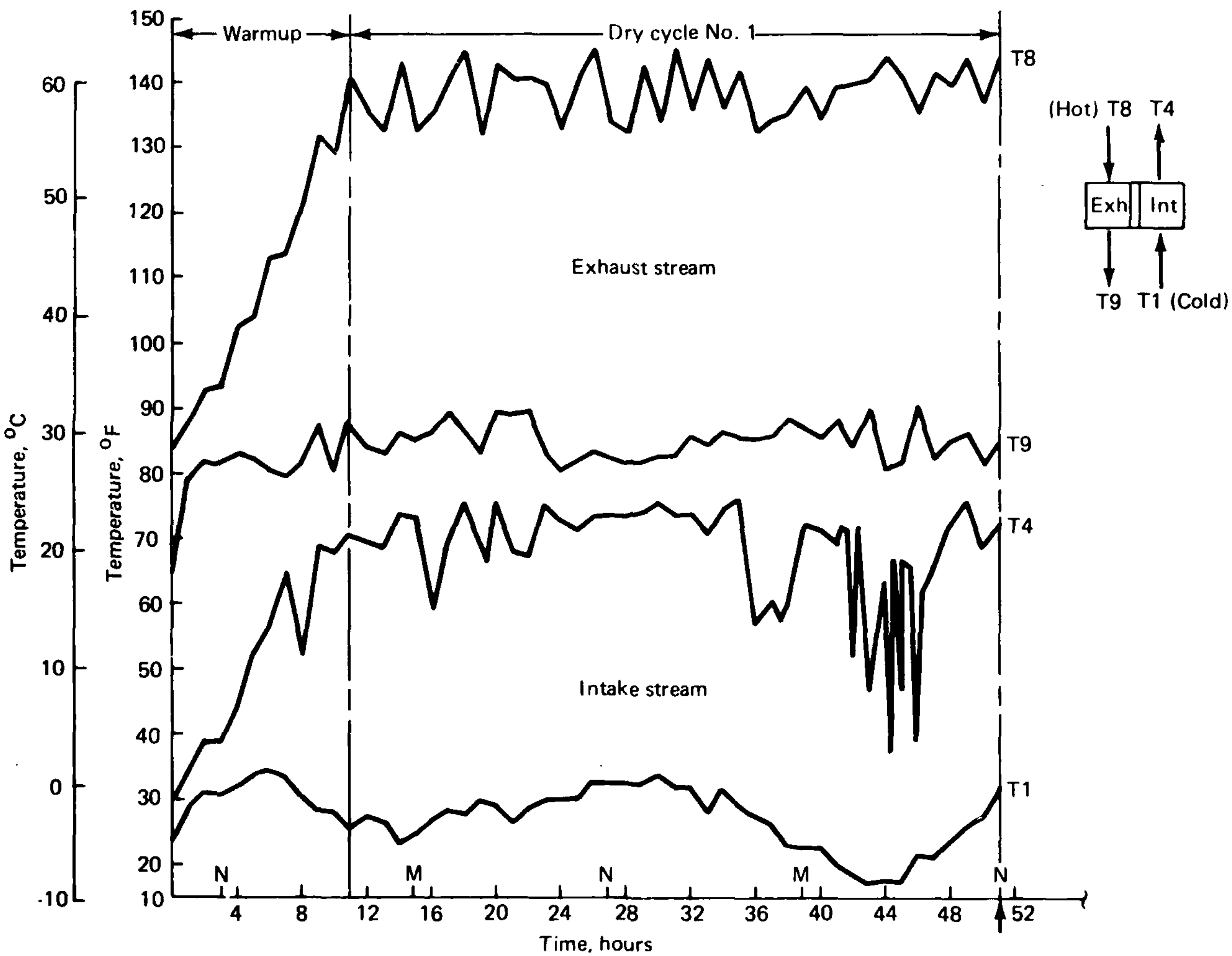
2717-039W

Fig. 19 Steam coil performance (standby)



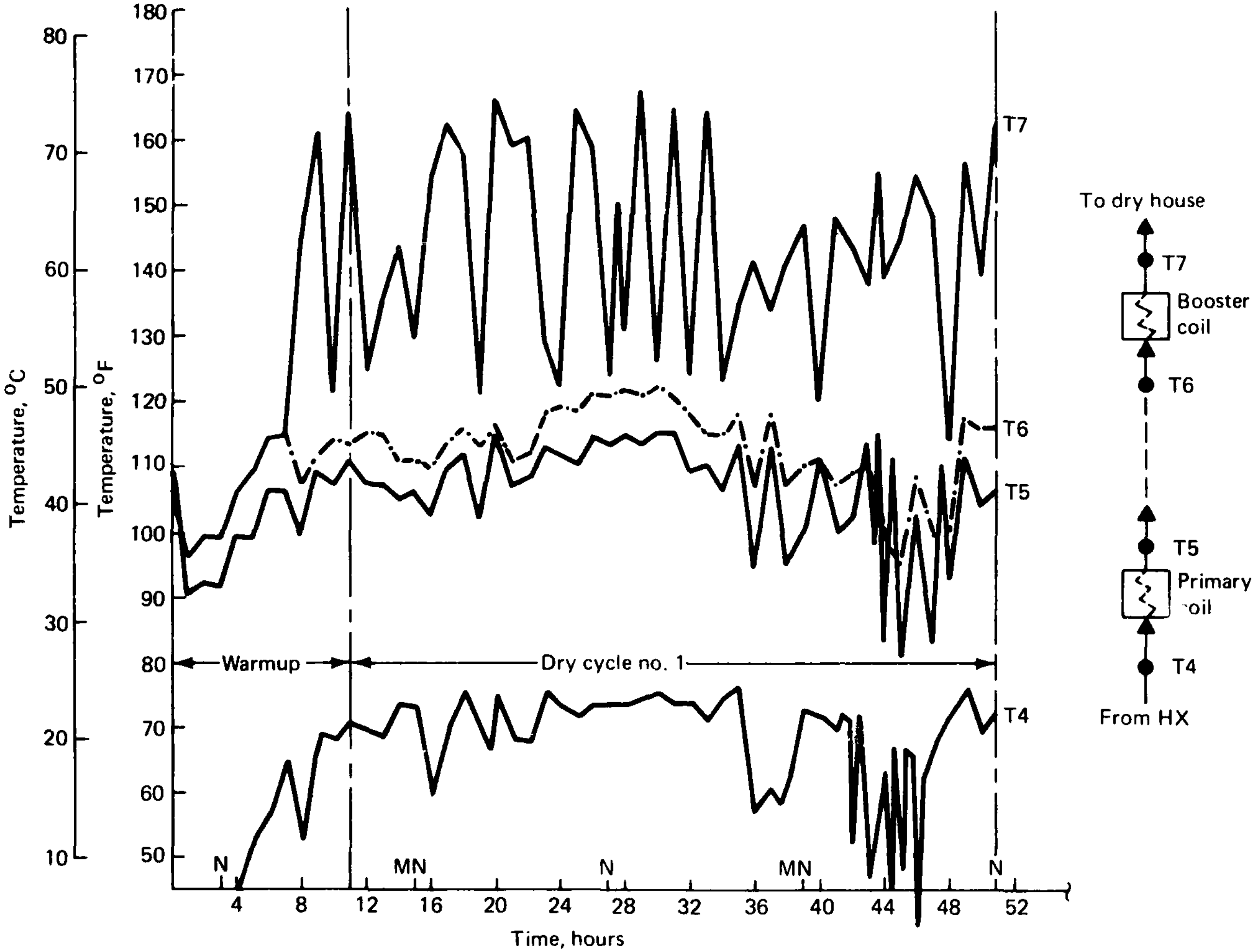
2717-040W

Fig. 20 Overall system performance (dry cycle no. 1)



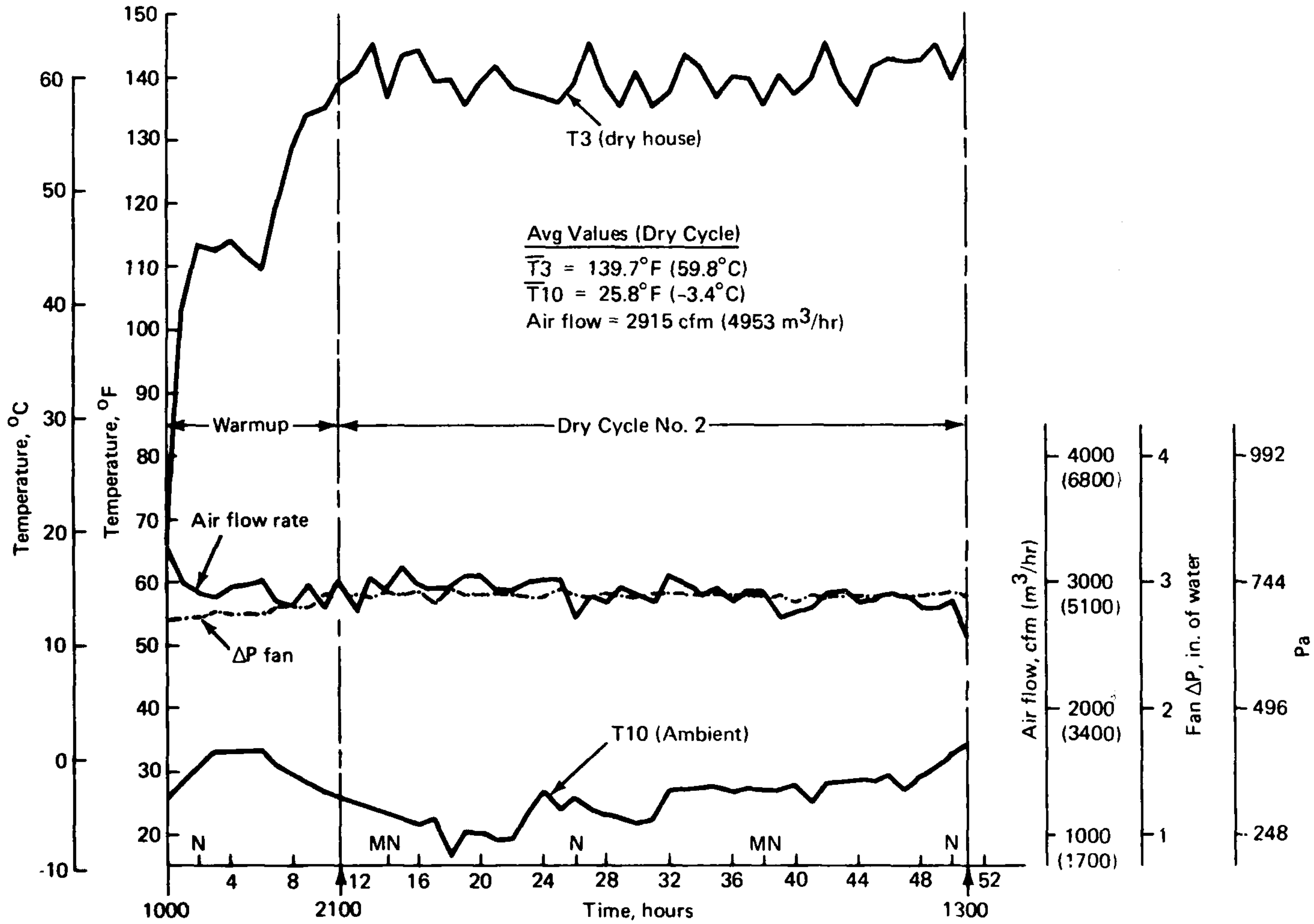
2717-041W

Fig. 21 Heat exchanger performance (dry cycle no. 1)



2717-042W

Fig. 22 Steam coil performance (dry cycle no. 1)



2717-043W

Fig. 23 Overall system performance (dry cycle no. 2)

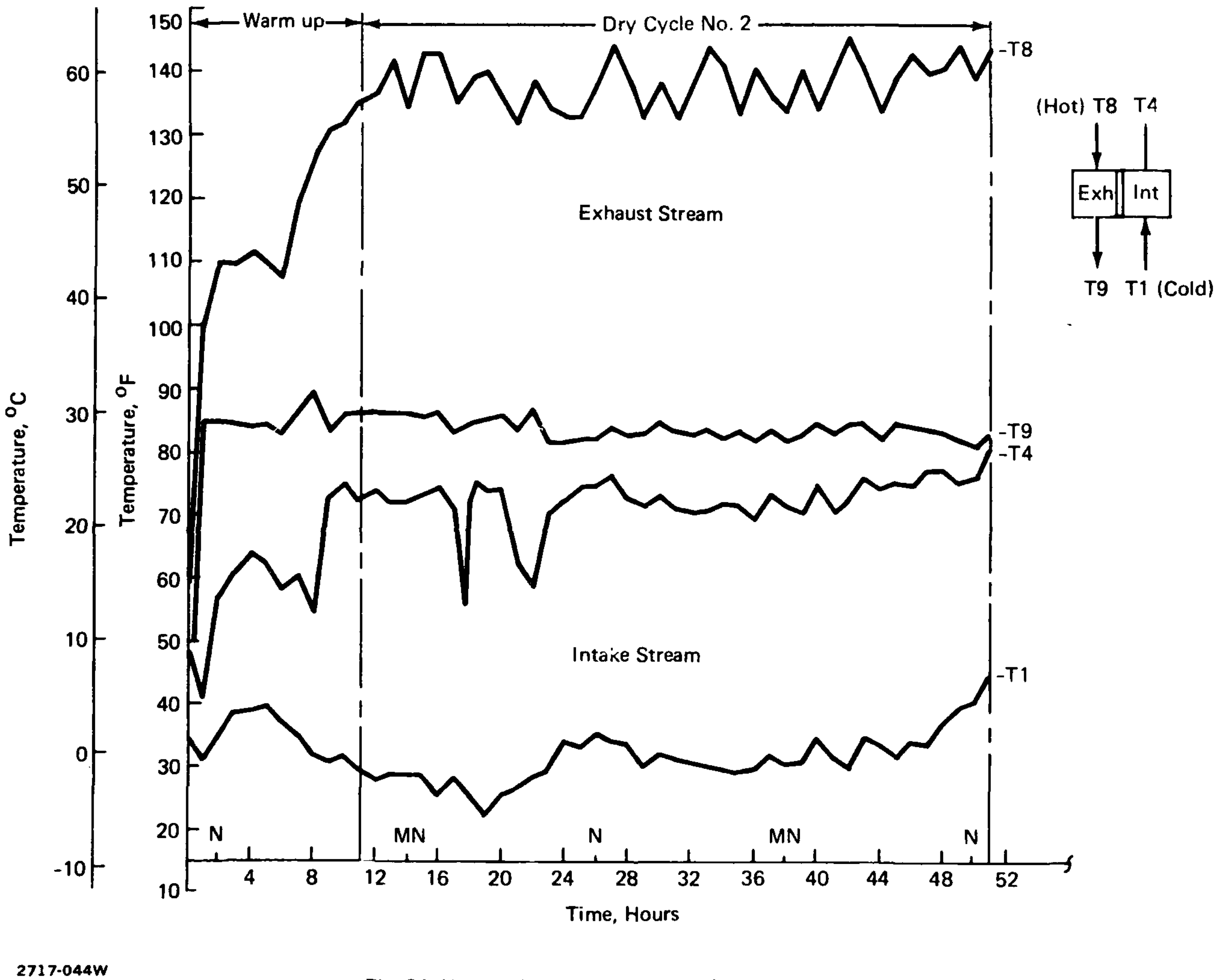
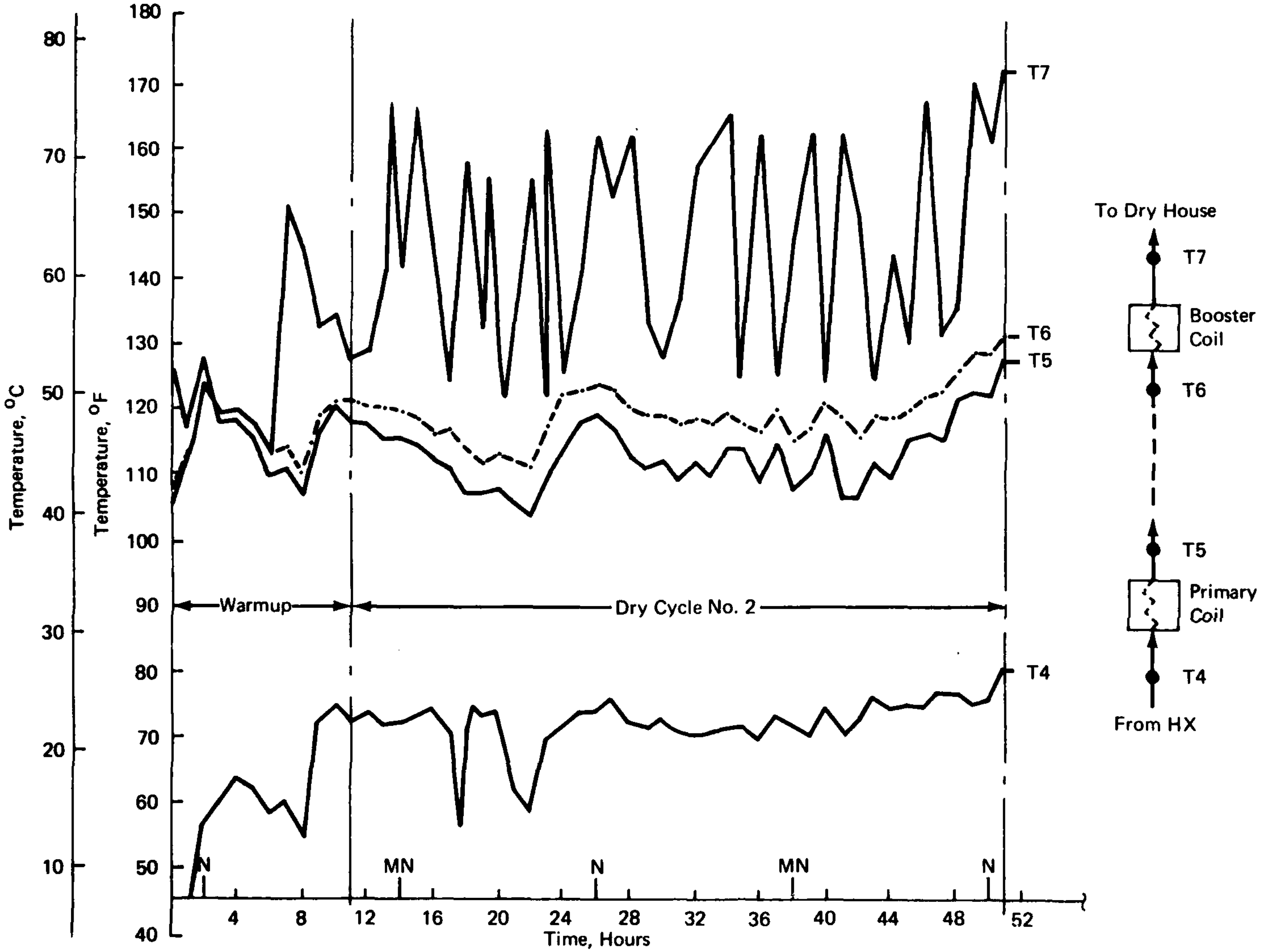
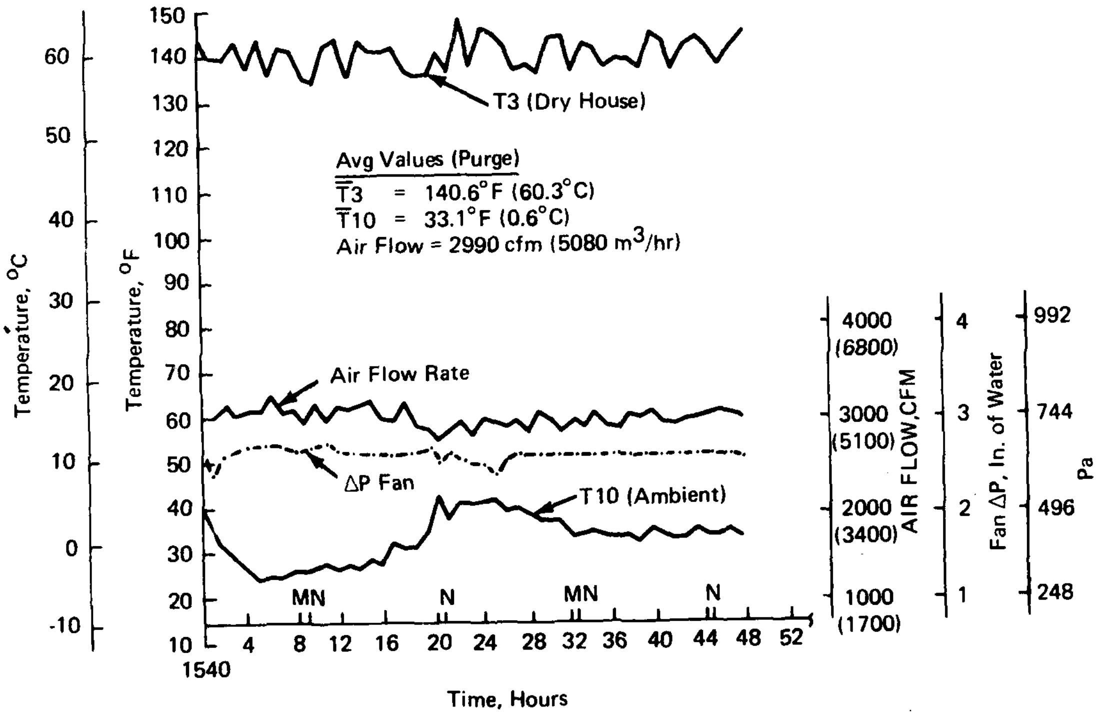


Fig. 24 Heat exchanger performance (dry cycle no. 2)



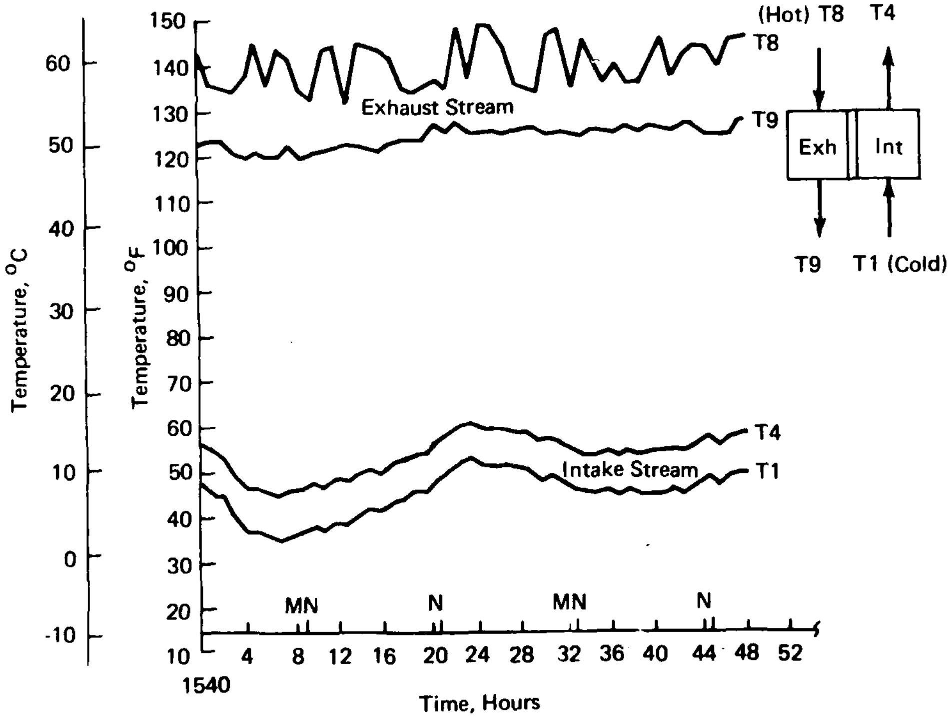
2717-045W

Fig. 25 Steam coil performance (dry cycle no. 2)



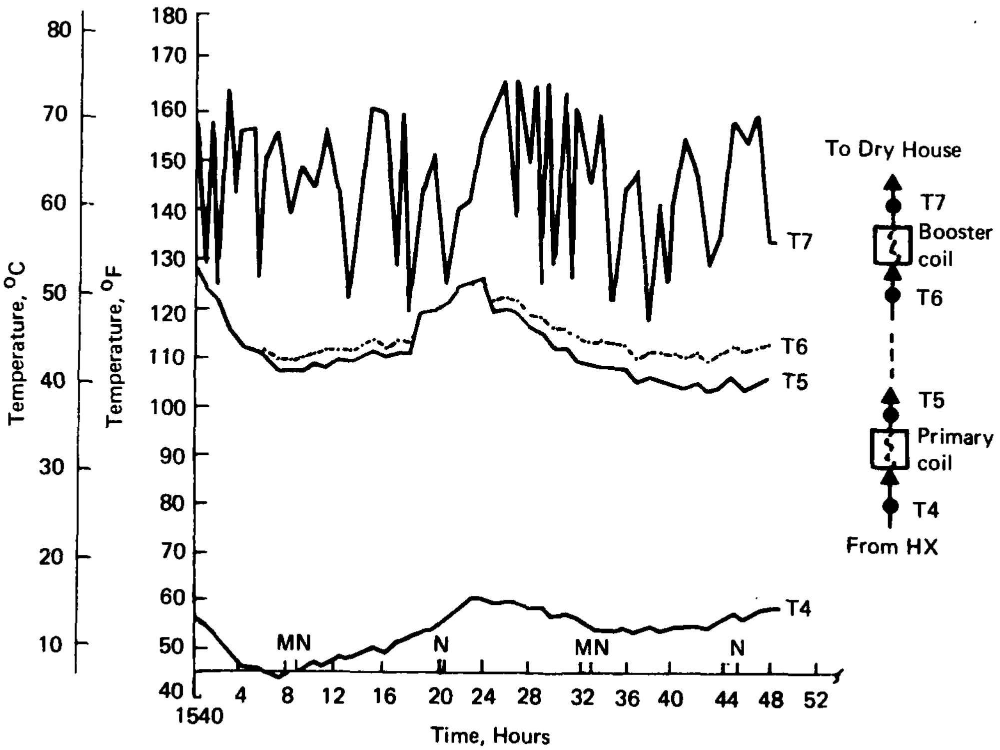
2717-046W

Fig. 26 Overall system performance (purge)



2717-047W

Fig. 27 Heat exchanger performance (purge)



2717-048W

Fig. 28 Steam coil performance (purge)

APPENDIX A

Calculation Procedures for Evaluating Waste Heat Recovery System

List of Symbols

A_D	-	Intake duct flow area, 2.88 ft^2 ($0.268m^2$)
C_p	-	Specific heat of air stream, 0.24 Btu/lb^{o}F (kw-hr/kg^{o}C)
E	-	Energy, Btu (kw-hr)
I	-	Current, amps
M_T	-	Total intake stream air flow, lb/hr (kg/hr)
M_x	-	HX exhaust airflow, lb/hr (kg/hr)
Q	-	Heat rate, Btu/hr (watts)
η	-	Overall energy recovery system effectiveness
ρ	-	Air density, lb/ft^3 (g/cc)
T	-	Temperature, ^{o}F (^{o}C)
τ_o	-	Initial time
τ_n	-	Time after n intervals
$\Delta\tau$	-	Time interval, hr
$\overline{V}$	-	Average duct velocity, ft/min (m/sec)
V_c	-	Centerline duct velocity, ft/min (m/sec)
(V/V_c)	-	D .ct correction factor = 0.9957
(KW-H)	-	Kilowatt-hour reading

Subscripts

T - Total

O - Initial time

n - Final time

EX - Exhaust side

I - Intake side

Calculations

1. Air flow rate (total intake stream):

   a) $\overline{V} = V_c \, (\overline{V}/V_C) \quad V_c$

   b) $M_T = \rho A_D \overline{V}$

   where ρ is evaluated at average temperature T5 using figure A-1

2. Exhaust stream flow rate (M_x):

   $$M_x = M_T \frac{(T4 - T1)}{(T8 - T9)}$$

3. Total system energy requirement (E_T):

   $$E_T = \int Q_T d\tau$$

   $$\approx M_T C_p \Sigma \; (T3 - T10)\Delta\tau$$

   for equal time increments

   $$E_T = M_T \, C_p \, \Delta\tau \, \Sigma \; (T3 - T10)$$

   where the quantity Σ (T3-T10) is the summation of the incremental temperature differences between the dry house and ambient as obtained from the paper tape data records. It can be computed as

   $$\Sigma \, (T3 - T10) = \Sigma T3_n - \tfrac{1}{2}(T3_o + T3_n) - \Sigma T10_n + \tfrac{1}{2}(T10_o + T10_n)$$

   which is the general form used to compute all other incremental temperature differences, with appropriate changes in inlet and outlet temperatures.

4. Energy recovered by heat exchanger system (E_{HX}):

   a) Intake stream

   $$E_{HX\,(I)} = M_T \, C_p \, \Delta\tau \Sigma \; (T4 - T10)$$

Note: The ambient temperature T10 is used as the low temperature reference instead of T1 because over one-half of the system pressure drop is attributable to the heat exchanger and its associated ducting. In addition, T10 is also the common reference for the total system requirement.

b) Exhaust stream:

$$E_{HX(EX)} = M_x C_p \Delta \tau \Sigma \ (T8 - T9)$$

for equal flow rates, $M_x = M_T$, and

$$E_{HX(EX)} = M_T C_p \Delta \tau \Sigma \ (T8 - T9)$$

c) Average HX recovery (equal flow rates):

$$E_{HX} = \tfrac{1}{2}(E_{HX(I)} + E_{HX(EX)})$$

5. Total energy from steam coils (E_s):

a) $E_s = M_T C_p \Delta \tau \Sigma \ (T7 - T4)$

or

b) $E_s = E_T - E_{HX}$

6. Energy input to HX fan motor (E_{Fan}):

$$E_{Fan} = (\text{kw-hr})\ (3413)$$

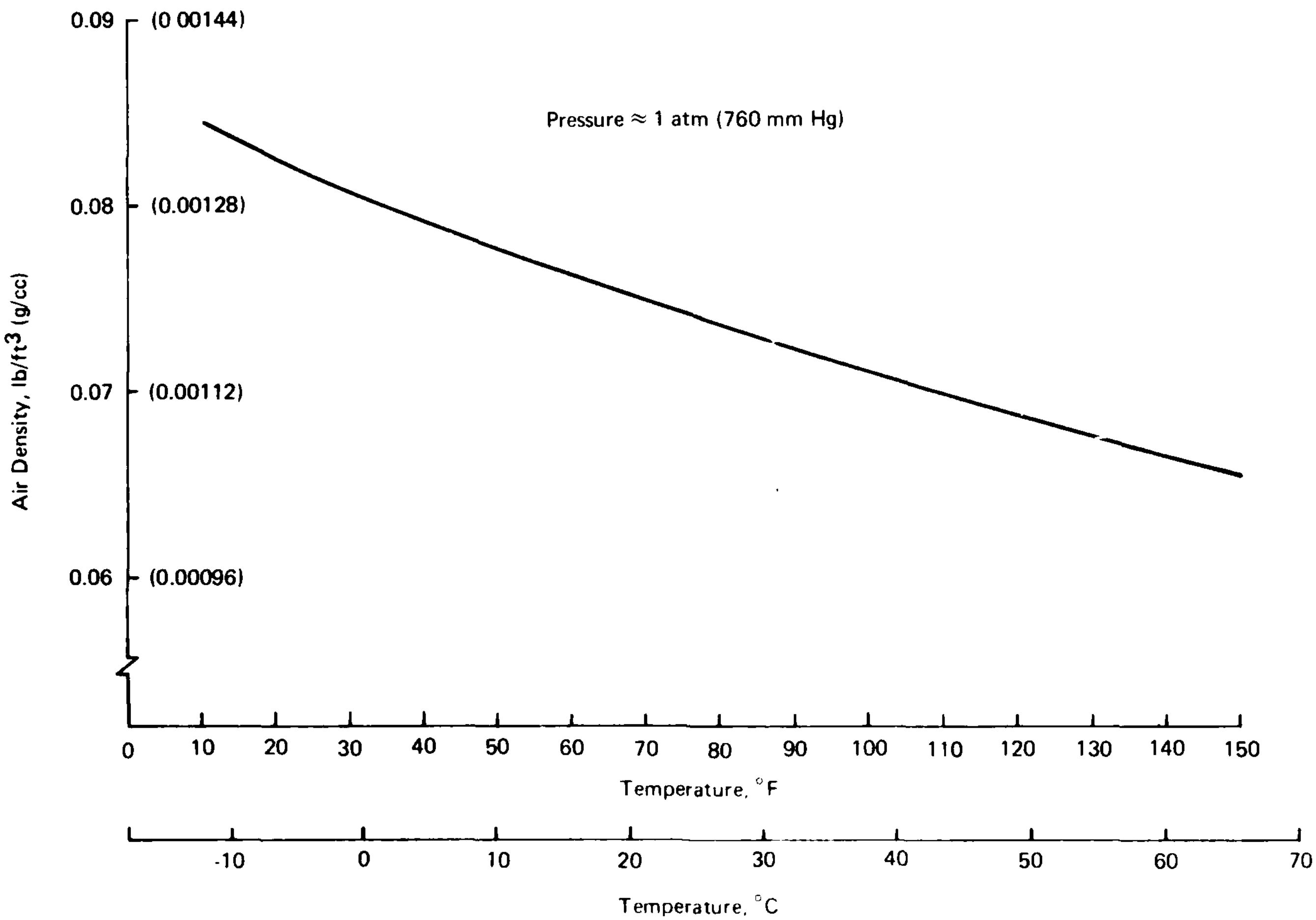
7. Overall energy recovery system effectiveness (η):

a) Based on averaged intake and exhaust stream values:

$$\eta = \frac{E_{HX}}{E_T}$$

b) Based on intake stream recovery:

$$\eta = \frac{E_{HX(I)}}{E_T}$$



2717-049

Fig. A-1 Air density vs temperature

APPENDIX B

Example of Economic Analysis

System: Existing dry house, no NG condensation permitted

cfm per system	= 5500
Number of systems	= 96

STATUS QUO

Recurring Costs

1. Steam used (per unit) = Heating rate x time

a) Dry cycle $= mCp\ (\Delta\ T)\ (\text{time})$

$$= (5500\ \text{cfm})\ (0.074\ \frac{\text{lb}}{\text{ft}^3})\ (0.024\ \frac{\text{Btu}}{\text{lb}^{\circ}\text{F}})$$

$$(60\ \frac{\text{min}}{\text{hr}})\ (140^{0}\text{F}-53^{\circ}\text{F})\ (7595\ \text{hr})$$

$$= 3872.69(10^6)\ \text{Btu}$$

b) Standby $= (5500)\ (0.074)\ (0.24)\ (60)\ (80-53)\ (1165)$

$$= 184.35(10^6)\ \text{Btu}$$

c) Total steam = Dry cycle + Standby

$$= 3872.69(10^6)\ \text{Btu} + 184.35(10^6)\ \text{Btu}$$

$$= 4057.04(10^6)\ \text{Btu per unit}$$

Latent heat at 40 psig = 919.6 Btu per lb

Total steam (lb) $= 4057.04(10^6) \times \frac{1}{919.6}$

$$= 4.412(10^6)\ \text{lb per unit}$$

2. Electricity used (per unit) at 5500 cfm

a) Dry cycle (kw-hr) $= (0.8\ \text{kw-hr per hr})\ (7595\ \text{hr})\ \frac{(5500)}{3000}$

$$= 11.139(10^3)$$

b) Standby (kw-hr) $= (2.4)(1165)\frac{(5500)}{3000} = 5.126(10^3)$

c) Total electricity $= 16.265(10^3)$ kw-hr per unit

3. Total energy costs = steam + electricity

a) Steam (1977) = \$3.80/1000 lb (4.412×10^6) (96 units)

= \$1.609(10^6)

b) Electricity (1977) = \$0.02531 (16.265) (10^3) (96)

= \$0.0395(10^6)

c) Total energy (1977) = \$1.649(10^6), for 96 systems

d) Energy Costs for 1980 = 1.05^3 (1977 Cost)

= 1.158 (\$1.649 x 10^6)

= \$1.909(10^6)

e) Costs escalate by 5% for each following year.

ALTERNATIVE

Non-Recurring Costs

Cost per system = \$24,250 + \$4800 = \$29,050

Total non-recurring cost = 96(\$29,050) = \$2789(10^6) for 96 systems

Recurring Costs

1. Steam used = status quo $(1-\bar{\eta})$

a) Dry cycle, $\bar{\eta}$ $= \frac{1}{4}(0.40) + 3/4(0.62) = 0.565$

Steam used $= 3872.69(10^6)(1-0.565)$

$= 1684.62(10^6)$ Btu $\times \frac{1}{919.6}$

$= 1832(10^6)$ lb per unit

b) Standby

Steam used = no recovery assumed, same as ...

$= 184.36(10^6)$ Btu $\times \frac{1}{919.6}$

$= 0.200(10^6)$ lb per unit

c) Total steam energy used = dry cycle + standby

$= 2.032(10^6)$ lb per unit

2. Total electricity used $\approx$ status quo

$= 16.265(10^3)$ kw-hr per unit

3. Total energy costs = steam + electricity

a) Steam (1977) $= \$3.80/1000$ lb (2.032×10^6) (96)

$= \$0.741(10^6)$

b) Electricity (1977) $= \$0.02531$ (16.265) (10^3) (96)

$= \$0.0395(10^6)$

c) Total energy (1977) $= \$0.781(10^6)$ for 96 systems

d) Total energy costs for 1980 $= 1.05^3(\$0.781)(10^6)$

$= \$0.904(10^6)$

e) Recurring costs escalate by 5% for each following year.

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