THE APPLICATION OF ENERGY CONSERVATION
TECHNIQUES IN AN INDUSTRIAL PLANT

by

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ABSTRACT

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An energy conservation program was developed in the fall of 1975 at General Extrusions, Inc. to maintain the operation of the plant at full production while staying within the natural gas allocation of 35,600,000 cubic feet per year. This program when fully implemented will minimize the use of propane as an alternate fuel. Before the energy conservation program, 8,000 gallons of liquid propane (720,000 cubic feet equivalent of natural gas) were used monthly to supplement the energy needs of the plant operation.

Group meetings were held with the plant personnel, stressing the importance of energy conservation in maintaining their jobs and seeking their help in this program. The energy conservation program then became a group effort with management and labor working jointly. A complete and detailed survey and study of all energy usage in the plant was done with the help of the maintenance staff and the operating supervisors. The survey and study was completed in two months and all the information and recommendations by the group were evaluated by a management committee.

The following problems were found during the energy usage study: the operation of the plant consisted of six days per week and twenty-four hours per day which left no flexibility in operating hours for cycling of

energy loads; the anodizing building required thirteen air changes per hour (consists of 157,000 CFM exhaust and 150,000 CFM fresh outside supply air) to purge the toxic fumes from the building continuously twenty-four hours per day, seven days per week to prevent corrosive attack on the equipment and finished product stored in the building; the 157,000 CFM exhaust consisted of low temperature though high humidity exhaust (80°F and 80% relative humidity); the make-up air heaters and the exhaust fans were positioned on the roof in a way that would make it difficult for the addition of heat recovery coils or wheels; even though the exhaust air went through wet scrubbers, most of it was too toxic or too humid to return to the plant; and finally, the first cost of installing heat recovery equipment was relatively high.

The following recommendations were implemented immediately:

1. The plant thermostat temperatures were set at 65°F and the employees agreed to dress accordingly,
2. Exhaust air from three tank immersion heaters was directed to another tank to supply 250°F temperature air for drying of finished extrusions (thereby eliminating the need for 200 CFH burner),
3. Inlet duct thermostats were installed in each of the three 50,000 CFM make-up air heaters to automatically turn off the burners when outdoor temperatures rose to 65°F (during unseasonable warm weather, even though the burners were on maximum turndown, 80°F air would be supplied to the plant, wasting as much as 500 CFH.)

Two additional recommendations of energy conservation systems were evaluated based on the following design constraints: first cost, operating and maintenance cost, quantity and cost of energy conserved, effect on in-plant air quality, and effect on employee environment.

Based on these evaluations, both systems were economically justified and it was determined that they would not degrade the in-plant air quality, therefore they were implemented. The following is a description of the two systems and a tabulation of the calculated design energy savings. The systems were fully operational on March 1, 1976.

The first energy conservation system was implemented to take advantage of the effectiveness of the exhaust fan-scrubber (20,800 CFM) that is handling air from the sulfuric acid tanks. This scrubber removes the contaminants from the exhaust air well below the allowable level for in-plant air designated by OSHA. Therefore, this heated cleaned air (65°F) was recycled back into the plant with the addition of a return duct system. This system reduces the make-up air requirements by 20,800 CFM and therefore saves the quantity of energy normally required to heat this amount of air from outside temperature to 65°F during the winter heating season. For the Youngstown area which has 6400 degree days per year and a plant operation of twenty-four hours per day six days per week, the quantity of energy conserved is estimated to be 3×10^9 BTU's annually.

The second energy conservation system that was implemented uses a heat pipe non-regenerative air-to-air heat recovery unit (Q-Dot) to recover energy from a 35,250 CFM exhaust fan (80°F and 80% relative humidity). This recovered energy is used to preheat 50,000 CFM of fresh outside supply air that is to be reheated to the 65°F required building temperature by a gas direct fired make-up air heater. The design conditions for this heat recovery system are: temperature of exhaust air is 80°F, temperature of supply air entering the coil is 37°F, temperature of supply air leaving the coil is 66°F and the recovery factor is .667. As can be seen from the design conditions, when the outside temperature is 37^{o}F or greater, no gas will be burned in the make-up air heater and 50,000 CFM will be supplied to the building at 65^{o}F or greater. If the outside temperature is less than 37^{o}F a greatly reduced quantity of gas will be burned to maintain the supply air at 65^{o}F. Computer modeling of this system was done to determine the optimum design and to calculate the estimated energy savings of 7 x 10^9 BTU's annually.

The energy required for the space heating of the anodizing building before the energy conservation program was implemented was 20 x 10^9 BTU's annually. Therefore, these two systems will save 10 x 10^9 BTU's annually or 50% of the space heating requirement of the building.

The two energy conservation systems have a first cost of $48,000.00 total. At current costs of $1.26 per MCF for natural gas and $.35 per gallon for propane, the energy savings of 10 billion BTU's is equivalent to approximately $36,000.00 (first 8.6 billion BTU's at propane costs and balance of savings at gas costs). Based on the above figures and the cost of money at 10%, the payback period for the total installation is approximately 1.5 years.

This thesis presents the detailed investigation, design and calculations of the two energy conservation systems that have been implemented. The present heating and ventilating system is described and alternate energy conservation and heat recovery systems are presented in detail. Computer modeling and economic analysis of the alternate systems is included.

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LIST OF SYMBOLS

SYMBOL	DEFINITION
BTU	British Thermal Unit (unit of energy)
CFH	Cubic Feet per Hour (natural gas)
CFM	Cubic Feet per Minute (air)
MCF	Thousand Cubic Feet (natural gas)

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CHAPTER I

INTRODUCTION

This thesis documents the study, design and implementation of the energy conservation program in the anodizing building at General Extrusions, Incorporated. General Extrusions is an aluminum extruding company with the capability of total finishing of an aluminum product including extruding, fabricating, heat treating, and anodizing. Their natural gas allocation was reduced to the 1968 level of consumption, less 10%. This allocation problem is even more critical since a considerable amount of natural gas fired equipment has been added since 1968. This necessitated the installation of a propane-air mix system to supplement the natural gas supply, thus providing the additional energy required but increasing the operating costs significantly. Note: at present energy prices, natural gas costs $1.26 per million BTU's thermal and propane costs $3.80 per million BTU's thermal.

The problem of energy shortages and increased cost has recently become one of the most difficult for plant management personnel throughout the country. At General Extrusions, this energy management program was implemented to try to make the gas allocation program more workable. The addition of propane as an alternate fuel to supplement the natural gas requirement has caused production costs to increase substantially and is therefore causing the plant to lose its competitive advantage in the market place. Thus, energy conservation became a must for this plant to maintain the competitive position in the market place that protects the employees' jobs and profitability of the operation and also to try to free some of the alternate fuel for increased production or equipment if required in future operation.

Production operations and space heating in the anodizing building (see Appendix pages 46 and 47 for description of anodizing process) require 95% of the present total natural gas - propane consumption of the plant, thus conservation in this part of the plant is of great importance. Because of the toxic fumes generated in the anodizing process, thirteen air changes per hour are required in the building continuously, twenty-four hours per day and seven days per week. Therefore, energy consumption of the process and space heat provide a fairly constant load and cannot be cycled or shut off periodically to save energy. The relatively high number of air changes per hour represents a very large energy requirement (20 million cubic feet of gas) during the winter to heat the outside make-up air to 65^{o}F, but is required because of the toxicity and corrosiveness of the fumes and the harm that they can cause to workers and finished product that is stored in the building.

This large volume of exhaust air (157,000 cubic feet per minute) does represent a source of waste heat, though this had been overlooked by other manufacturers of heat recovery equipment and engineering firms because of the relatively low exhaust temperatures (80^{o}F maximum). The unique feature of this energy conservation system is the application of the heat recovery equipment to a relatively low temperature but very high humidity (80% relative humidity) exhaust.

The following chapters will present the systems approach to the problem of energy conservation applied to the heating and ventilating system in the anodizing building at General Extrusions.

CHAPTER II

PROBLEM DESCRIPTION

Heating and Ventilating System Operation

This section of the thesis will describe the operation of the existing heating and ventilation system including a description of all the components in the system, a description of a model of the thermal system of the building and the design constraints of the system. The ventilation system consists of three exhaust fan-separator wet scrubber units of 52,700 CFM, 48,400 CFM, and 20,800 CFM capacity respectively and an exhaust fan of 35,250 CFM capacity. The heated make-up supply air is provided by three 50,000 CFM capacity make-up air heaters. See Appendix, page 48 for a plan view of the building. See Figure 1 for a diagram of the equipment and the air flow direction of the heated supply air.

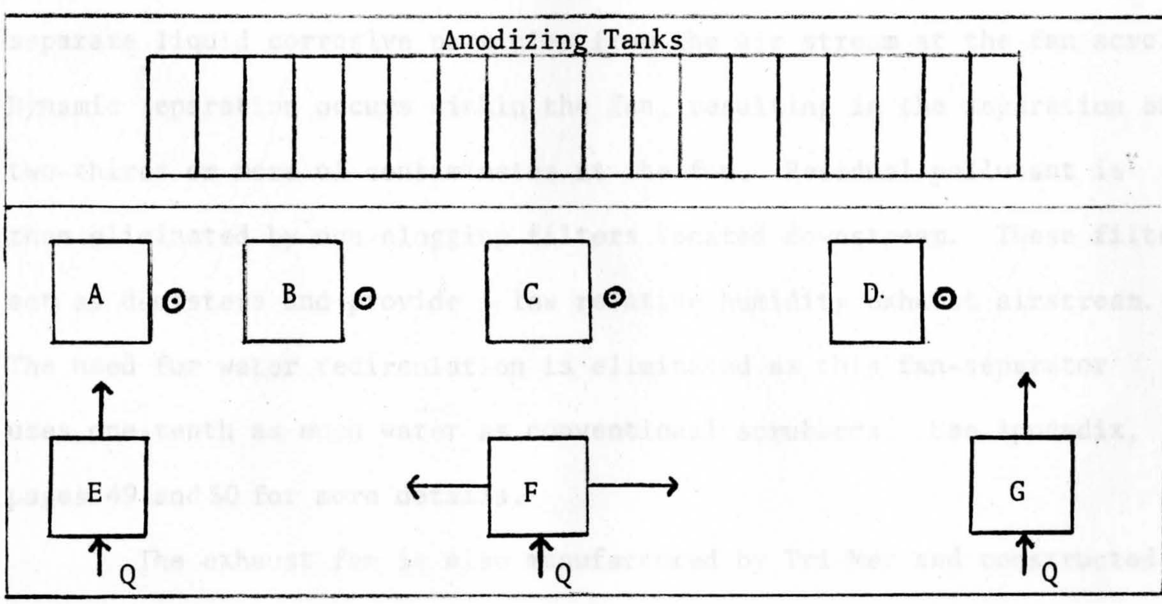


Fig. 1. Equipment Diagram

TABLE 1

HEATING AND VENTILATING EQUIPMENT

Mark	Description	Output Air Temperature
A	48,400 CFM exhaust fan-separator	$74^{\circ}F$
B	52,700 CFM exhaust fan-separator	$74^{\circ}F$
C	20,800 CFM exhaust fan-separator	$66^{\circ}F$
D	35,250 CFM exhaust fan	$80^{\circ}F$
E	50,000 CFM make-up air heater	$65^{\circ}F$
F	50,000 CFM make-up air heater	$65^{\circ}F$
G	50,000 CFM make-up air heater	$65^{\circ}F$

Description of Exhaust Fan and Fan-Separator

The exhaust fan-separator is a patented unit manufactured by Tri-Mer Corporation of Owosso, Michigan. It is constructed of reinforced rigid unplasticized PVC (Polyvinyl chloride) to provide chemical corrosion resistance. The design of the unit utilizes a centrifugal fan to separate liquid corrosive particles from the air stream at the fan scroll. Dynamic separation occurs within the fan, resulting in the separation of two-thirds or more of contaminates at the fan. Residual pollutant is then eliminated by non-clogging filters located downstream. These filters act as demisters and provide a low relative humidity exhaust airstream. The need for water recirculation is eliminated as this fan-separator uses one-tenth as much water as conventional scrubbers. See Appendix, pages 49 and 50 for more details.

The exhaust fan is also manufactured by Tri-Mer and constructed of PVC. See pages 51-52 in Appendix for more details on the fan and fan tables.

The exhaust fan-separator, Mark A (see Figure 1), exhausts fumes from the following four tanks: alkaline clean at 160°F; double rinse at ambient temperature; nitric acid clean at 170°F and etch at 140°F. The exhaust fan-separator, Mark B, exhausts fumes from three rinse tanks at ambient temperature and one bright dip tank at 200°F. The exhaust fan-separator, Mark C, exhausts fumes from two sulfuric acid anodizing tanks at ambient temperature. The exhaust fan, Mark D, exhausts fumes from the following four tanks: nickel acetate at 160°F; double rinse at ambient temperature; and two hot water seal at 200°F.

Description of Make-up Air Heater

The three make-up air heaters are identical and produce 50,000 CFM of air heated from outside temperature to 65°F. The units are Model MA50 manufactured by W. C. Grant Company and are direct gas fired with a rated efficiency of 92%. These units are capable of producing an 80° temperature rise with a 4,400,000 BTU per hour burner. The units are thermostatically controlled from three set points. There is a thermostat in the inlet duct and the outlet duct of the make-up air heater and also one in the building. If the inlet duct thermostat registers a temperature of 65°F or above, the main burner stays off. If the thermostat in the building registers below 65°F, the main burner comes on and requires the duct outlet temperature to be above 67°F.

The Grant make-up heaters are an efficient way to supply heated, outside air to replace in-plant air lost by combustion processes and exhaust ventilation. Instead of permitting this lost air to be replaced through uncontrolled openings, and heated by inefficient mixing, the unit introduces and efficiently heats fresh air, controlling both temperature and distribution. During summer operation the burner is turned off and the fan operates independently. This insures proper exhaust fan operation and proper natural stack operation. See Appendix, pages 53 to55 for more details and fan data.

Thermal System Model

A model of the thermal system of the anodizing building will be developed in the following paragraphs. The building conductive heat loss calculation will be presented and then other heat inputs and outputs will be discussed.

For all of the building sections a sketch of the wall section will be provided and the heat loss will be shown in equation form for each view and also for the total building. The following equation will be used for calculation of transmission heat loss:

$$\text{Heat loss} = U \times (T_i - T_o) \times A \qquad (1)$$

where: Heat loss = the BTU per hour transmission loss from inside air to outside air

U = overall heat transmission factor (BTU/hr ft^2 F^o) dependent on building material

T_i = inside air temperature (F^o)

T_o = outside air temperature (F^o)

A = square feet of building wall area (ft^2)

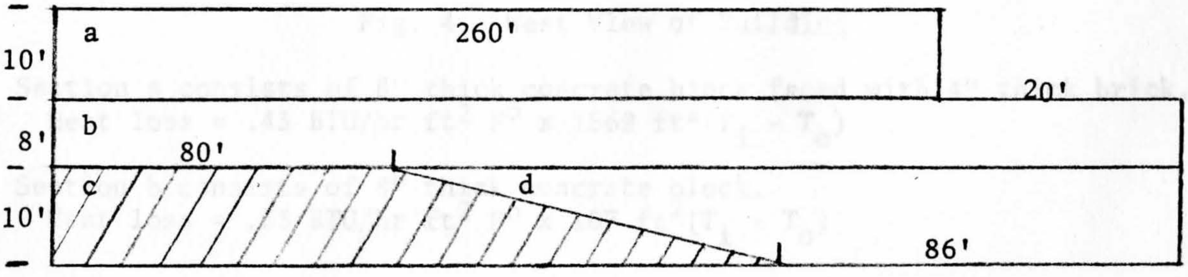


Fig. 2. South View of Building

Section a consists of 1/16" thick corrugated fiberglass sheeting.
Heat loss = 1.5 BTU/hr ft^2 F^o x 2600 ft^2(T_i - T_o)

Section b consists of 8" thick concrete block faced with 4" thick brick.
Heat loss = .43 BTU/hr ft^2 F^o x 2240 ft^2(T_i - T_o)

Section c consists of 8" thick concrete block below grade level.
Heat loss = 4 BTU/hr ft^2 x 1370 ft^2

Section d consists of 8" thick concrete block.
Heat loss = .53 BTU/hr ft^2 F^o x 1430 ft^2(T_i - T_o)

Total Heat Loss for South View:
Heat loss$_s$ = 5621 BTU/hr F^o(T_i - T_o) + 5480 BTU/hr

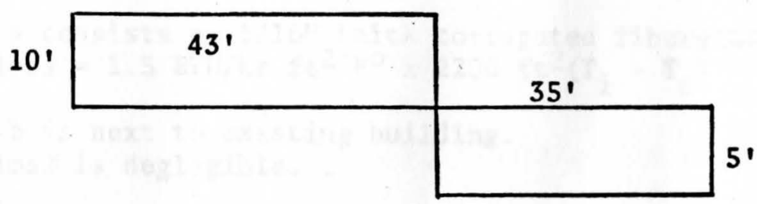


Fig. 3. East View of Building

Section consists of 8" thick concrete block.
Heat loss = .53 BTU/hr ft^2 F^o x 605 ft^2(T_i - T_o)

Total Heat Loss for East View:
Heat loss$_e$ = 321 BTU/hr F^o(T_i - T_o)

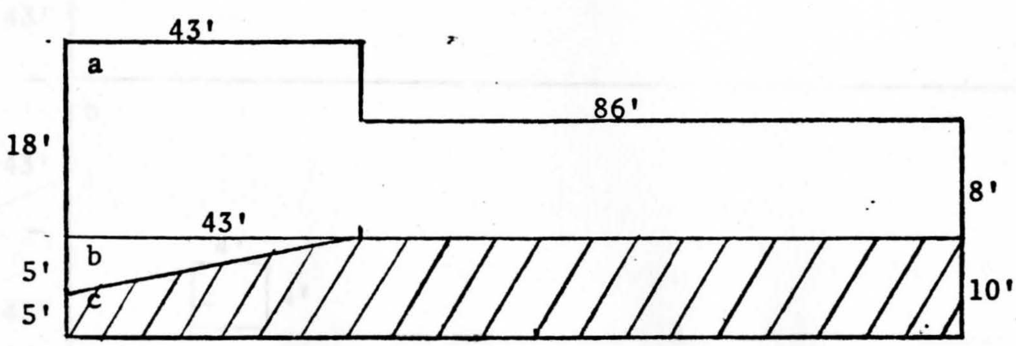


Fig. 4. West View of Building

Section a consists of 8" thick concrete block faced with 4" thick brick.
Heat loss = .43 BTU/hr ft^2 F^o x 1569 ft^2(T_i - T_o)

Section b consists of 8" thick concrete block.
Heat loss = .53 BTU/hr ft^2 F^o x 107 ft^2(T_i - T_o)

Section c consists of 8" thick concrete block below grade level.
Heat loss = 4 BTU/hr ft^2 x 1183 ft^2

Total Heat Loss for West View:
Heat $loss_w$ = 731 BTU/hr $F^O(T_i - T_o)$ + 4732 BTU/hr

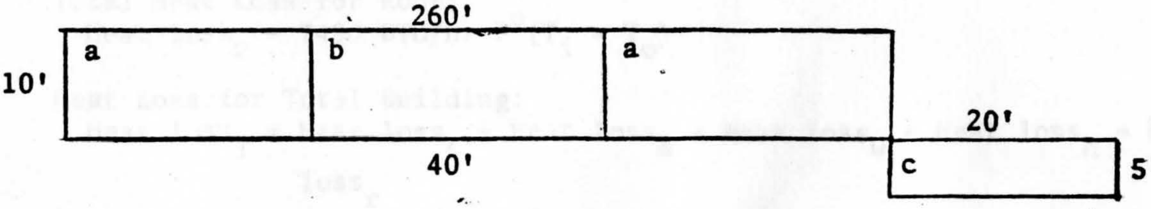


Fig. 5. North View of Building

Section a consists of 1/16" thick corrugated fiberglass sheeting.
Heat loss = 1.5 BTU/hr ft^2 F^O x 2200 $ft^2(T_i - T_o)$

Section b is next to existing building.
Heat loss is negligible.

Section c consist of 8" thick concrete block.
Heat loss = .53 BTU/hr ft^2 F^O x 100 $ft^2(T_i - T_o)$

Total Heat Loss for North View:
Heat $loss_n$ = 3353 BTU/hr $F^O(T_i - T_o)$

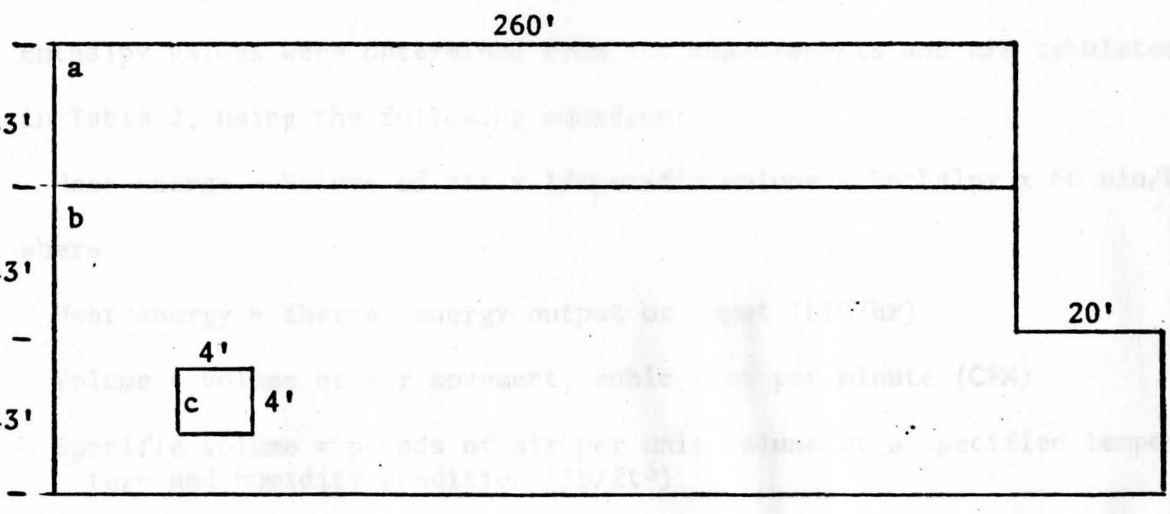


Fig. 6. Roof of Building

Section a consists of built up roofing 1" rigid insulation with 2" wood decking.
Heat loss = .168 BTU/hr ft^2 F^O x 11180 $ft^2(T_i - T_o)$

Section b consists of built up roofing 1" rigid insulation with metal decking.

Heat loss = .232 BTU/hr ft^2 F^o x 22980 $ft^2(T_i - T_o)$

Section c consists of 15 individual single pane skylights.

Heat loss = 1.13 BTU/hr ft^2 F^o x 240 $ft^2(T_i - T_o)$

Total Heat Loss for Roof:

Heat $loss_r$ = 7480 BTU/hr $F^o(T_i - T_o)$

Heat Loss for Total Building:

$$\text{Heat loss}_T = \text{Heat loss}_s + \text{Heat loss}_e + \text{Heat loss}_w + \text{Heat loss}_n + \text{Heat loss}_r$$

$$= 17506 \text{ BTU/hr } F^o(T_i - T_o) + 10212 \text{ BTU/hr}$$

<u>Tabulation of Thermal Output and Input</u>

The space in the anodizing building will be considered as the thermal system. The major source of heat output (loss) from the thermal system of the building consists of the exhaust of heated contaminated air. The quantity of heat output will be calculated from temperature and humidity measurements that were made during operation of the system. Enthalpy values were determined from the measurements and are tabulated in Table 2, using the following equation:

Heat energy = Volume of air x 1/specific volume x Enthalpy x 60 min/hr (2)

where:

Heat energy = thermal energy output or input (BTU/hr)

Volume = volume of air movement, cubic feet per minute (CFM)

Specific volume = pounds of air per unit volume at a specified temperature and humidity condition (lb/ft^3)

Enthalpy = a thermodynamic property which represents the measure of heat energy in a system above some datum temperature, the quantity of energy in one pound of air and the water vapor in it for a specified temperature and humidity condition (BTU/lb)

TABLE 2

DESCRIPTION OF EXHAUST

Description	Volume (CFM)	Conditions	Enthalpy (BTU/1b)	Heat Output (BTU/hr)
Exhaust #1	48,400	74°F 70%RH	31.6	6,698,000
Exhaust #2	52,700	74°F 70%RH	31.6	7,293,000
Exhaust #3	20,800	66°F 60%RH	25.0	2,328,000
Exhaust #4	35,250	80°F 80%RH	38.8	5,862,000
			Total	22,181,000

The anodizing building contains thirty-one process tanks, fourteen of which are heated. See Appendix, page 48 for plan view of tank layout. The heat loss from the tanks is tabulated in the Appendix on pages 56 and 57. Total heat loss from the tanks is 3,156,000 BTU/hr. By the definition of our thermal system model this is considered as a heat input to the building.

The make-up air to the plant consists of 150,000 CFM of outside fresh air heated to conditions of 65°F and 60%RH. Using the above formula and determining the enthalpy value of 24.3 BTU/1b for these conditions, this represents a heat input of 16,321,000 BTU/hr.

The tanks are heated by gas fired immersion burners. Eleven of these are vented to the outside of the building. Three of these are vented to a drying tank in the building and all of this escapes to the inside of the building. This represents a heat input of 2,396,000 BTU/hr.

Miscellaneous heat inputs consist of lighting, 95.35 kw, which is equivalent to 326,000 BTU/hr, and air infiltration from other buildings and outside of the buildings. A constant negative pressure is maintained in the anodizing building to localize the contaminated air (157,150 CFM exhaust and 150,000 CFM supply). This 7150 CFM difference represents a heat input which is based on average winter conditions. It is assumed that 3,575 CFM ($30^{o}F$ 50%RH) infiltrated from outdoors through cracks in the walls of the building and loose fitting doors and windows and 3,575 CFM ($65^{o}F$ 60%RH) infiltrated through open doorways of connected buildings. The following calculation based on equation (2), page 9, represents the average heat input due to this infiltration:

$$\text{Heat input} = [3575 \text{ CFM } (30^{o}\text{F, 50\%RH}) \times .080 \text{ lb/ft}^3 \times 5 \text{ BTU/lb} \times 60\text{min/hr}]$$
$$+ [3575 \text{ CFM } (65^{o}\text{F, 60\%RH}) \times .075 \text{ lb/ft}^3 \times 24.4 \text{ BTU/lb} \times 60 \text{ min/hr}]$$
$$= 477{,}000 \text{ BTU/hr for average winter design conditions}$$

Values of heat input and output will be tabulated in Table 3 for an average winter temperature of $30^{o}F$.

TABLE 3

VALUES OF HEAT INPUT AND OUTPUT

Heat Input		Heat Output	
Description	Quantity (BTU/hr)	Description	Quantity (BTU/hr)
Make-up Air	16,321,000	Exhaust	22,181,000
Tank loss	3,156,000	Building loss	623,000
Lighting	326,000		
Infiltration	477,000		
Tank Exhaust	2,396,000		
	22,676,000 BTU/hr Total		22,804,000 BTU/hr Total

As can be seen from Table 3 the total heat input is approximately equal to the total heat output. The calculated values differ by less than 1%. For values this large and due to the averaging done in the calculations, the values are considered equal and the temperature of the thermal system should remain fairly constant.

Economic Analysis of Additional Insulation

An analysis of the addition of a urethane foam insulation sprayed on the fiberglass sheeting in the anodizing building has been done. The area of largest heat conduction loss in the building consisted of the north and south walls of the building (the corrugated fiberglass sheeting). This could be foam insulated from outside of the building and sprayed with a weather resistant coating. Cost of the material and installation would be \$1.00/square foot. Total area was 4,800 square feet and would represent an added cost of \$4,800.00.

The added insulation would decrease the U factor for the section to .24 BTU/hr ft^2 F^o. This would represent a savings of heat energy of 212,000 BTU/hr for average winter temperature of 30^oF. [$Q = (1.5 - .24)$ BTU/hr ft^2 F^o x 4,800 ft^2 x $(65^o - 30^o)$]. Therefore, with the added insulation, an additional 212,000 BTU's would be exhausted every hour. The added quantity of heat through the recovery system would be 48,000 BTU's per hour [(35,250/157,150) x 212,000 BTU]. Based on a recovery efficiency of .66 (to be explained later), this represents a savings of 32,000 BTU/hr.

For a five month continuous operating period the estimated average savings would be 1.15×10^8 BTU (32,000 BTU/hr x 24 hr/day x 30 day/mo. x 5 mo.). This is an equivalent of 128 MCF of natural gas based on 90% conbustion efficiency. This is equivalent to \$161.00 per year based on present costs of natural gas (\$1.26/MCF). Therefore the additional insulation would require twenty-nine years to pay for itself based on simple payback with no interest. This is not cost effective.

Design Constraints

The two main constraints in the systems approach to the problem of energy conservation in the anodizing building are capability of the system to meet OSHA requirements for concentration levels of toxic fumes in the plant and cost effectiveness of the system based on life cycle costing.

Other design constraints would apply to existing components in the heating and ventilating system. Basically the adaptability of the existing fans would be investigated. This would involve the ability of the fan to operate with increased loads due to additional static and velocity pressure and higher speeds.

CHAPTER III

INVESTIGATION OF POSSIBLE ENERGY CONSERVATION SYSTEMS

There are three major areas of study to help conserve energy in the anodizing building. They are: reduction in the total volume of air handled; recirculation of warm uncontaminated air from processes; recovery of heat energy from the exhaust air. These areas of study will now be presented.

Reduction of Air Volume

A reduction in the total air volume handled requires careful engineering study of exhaust system design. There is no rule of thumb which permits the arbitrary cutting of exhaust volumes by a certain percent. Such an approach cannot be justified on the basis of hood and system design. A reduction of total air volume handled can be accomplished by conducting a careful inventory of all exhaust and supply systems in the plant, determining which are necessary, which can be replaced with more efficient systems or hood designs and which systems may be obsolete.

Exhaust volume must be of a sufficient level to maintain quality in-plant air. Proper design of exhaust hoods is necessary if a local exhaust system is to effectively control contamination at its source with a minimum air flow and power consumption. The theory of capture velocity depends on the creation of air flow past the source of contaminant sufficient to remove the highly contaminated air around the source or issuing from that source. Basically, hood design requires sufficient knowledge of a process or operation so the most effective hood or enclosure can be installed to provide minimum exhaust volumes for effective contaminant control.

The Tri-Mer Corporation designed and installed the present exhaust system based on years of experience in control of air pollution and on in-plant air conditions established by OSHA codes through the American Conference of Governmental Industrial Hygienists.

The actual volume of exhaust was then calculated to provide in-plant air conditions with toxic contaminants below the Threshold Limit Values (TLV). These values are for air borne toxic materials which are to be used as guides in the control of health hazards and represent time weighted concentrations to which nearly all workers may be exposed eight hours per day over extended periods of time without adverse effects. TLV refer to time weighted concentrations for a 7 or 8 hour workday and 40 hour workweek. They should be used as guides in the control of health hazards and should not be used as fine lines between safe and dangerous concentrations.

Threshold Limit Values are based on the best available information from industrial experience, from experimental human and animal studies, and, when possible, from a combination of the three. The basis on which the values are established may differ from substance to substance; protection against impairment of health may be a guiding factor for some, whereas reasonable freedom from irritation, narcosis, nuisance or other forms of stress may form the basis for others. The values have been developed by the American Conference of Governmental Industrial Hygienists and can be found in the Industrial Ventilation Manual.

It was determined from a review of the existing exhaust-ventilation system of the plant with Tri-Mer and a consultation with representatives of OSHA and General Extrusions that the quality of in-plant air would be affected adversely if there was an appreciable reduction in exhaust volume. Not only would this affect the plant workers but the acidic fumes could affect the finish on the aluminum extrusions that are warehoused in the anodizing building. This would be a costly problem that would produce scrap products. Thus, no energy savings could be produced in this area of study.

Recirculation of Warm Uncontaminated Air from the Process

Acceptance of a system that will clean the exhausted air and recirculate it into the building will depend on the degree of health hazard associated with the particular contaminant or toxic gas being exhausted, as well as on other factors discussed below. Factors to be considered are:

1. It is usually considered necessary to provide general ventilation air in addition to that recirculated so that there is, in effect, continuous dilution of any recirculated contaminants. If all the supply ventilation air is to be recirculated air, care must be taken to evaluate all the possible contaminants, not just the major contaminants normally concerned.
2. Recirculation systems should be designed to bypass to the outdoors, rather than to recirculate, when weather conditions permit. If a system is intended to conserve heat in winter months and if adequate "make-up" air is available, the system can discharge outdoors in warm weather.

3. Wet scrubbers also act as humidifiers. Recirculation of humid air from such equipment could cause high humidity and condensation problems. Additional ventilation should be provided to prevent excess humidity.
4. The layout and design of the recirculation ductwork should provide adequate mixing with other supply air and avoid uncomfortable drafts on workers or air currents which would upset the capture velocity of local exhaust hoods.
5. Routine testing, maintenance procedures, and records should be developed for the recirculating system.

The exhaust air from the three exhaust fan-separators and the one exhaust fan was studied to determine if any of this air could be returned directly to the plant. It was determined that exhaust fan-separators, Marks A and B (see Figure 1), do not remove enough of the toxic contaminants from the exhaust air to allow the recirculation of this air. This is based on recent tests of the toxicity of the exhaust air by Tri-Mer and a comparison to the TLV values. The exhaust fan, Mark D (see Figure 1), exhaust a very high humidity (80%RH) air and for this reason could not be recirculated into the building. The exhaust fan-separator, Mark C (see Figure 1), exhausts from the sulfuric acid tanks. Since sulfuric acid is hygroscopic and this type of separator is very efficient, it was determined that this system should be investigated further.

After reviewing the air quality tests of the exhaust from this fan-separator, reviewing the Threshold Limit Value for sulfuric acid, and discussing this application with Tri-Mer (the fan-separator manufacturer), it was determined that this air could be recirculated without adversely affecting the quality of in-plant air.

Threshold Limit Value for sulfuric acid is 1 milligram per cubic meter of air. Tests of the exhaust showed a value of .1 milligram per cubic meter of air. Tri-Mer will warranty that any air from this fan-separator that is returned to the plant will be acceptable for in-plant conditions as established by OSHA codes through the American Conference of Governmental Industrial Hygienists. Since this system is presently providing a ten to one factor of safety, it was determined to return this exhaust air directly into the plant.

This recirculation system would return 20,800 CFM of ambient air into the plant after the sulfuric acid was removed. This will reduce the "make-up" air requirements by 20,800 CFM, therefore this volume of outside air will not have to be heated to the 65^{o}F in-plant temperature. After discussion of this application with the bidding contractors, it was determined that because the sulfuric acid concentration is so low (.1 milligram per cubic meter) in the exhaust air, the ductwork material could be galvanized sheet metal and still provide a long maintenance free life.

Requests for quotation on this return air system were given to three sheet metal contractors. The following bids were received:

1. Thompson Heating and Cooling Company - $2,053.00 for a 20 gauge sheet metal installation.
2. A.A. Samuels Sheet Metal Company - $3,308.00 for a 20 gauge sheet metal installation.
3. Woodward Heating and Air Conditioning Company - $4,186.00 for an 18 gauge sheet metal installation.

4. Woodward Heating and Air Conditioning Company - \$7,890.00 for a 1/4 inch thick PVC installation.

The contractor recommended for the system was Woodward Heating and Air Conditioning Company. They were recommended because of their experience in this field, their fine installation on the existing system and the best proposal in terms of material and design. They were awarded the contract.

This system also required new v-belt drives for two of the Grant Make-up Air Heaters to reduce their volume to 39,600 CFM each. An adjustable pitch sheave was required for one unit and a smaller fixed pitch sheave was required for the other unit. The same length v-belts can be used. Cost for the new sheaves was \$157.72. Therefore, the total cost of the recommended system was \$4,343.72. The annual energy dollar saving will be calculated and tabulated in Chapter 4.

This system will require no change in plant operation since the exhaust fans are continuously operating twenty-four hours per day and seven days per week. A manual bypass damper has been incorporated in the design to allow all of the air to be exhausted outside if the fan-separator does not function properly.

Recovery of Heat Energy from the Exhaust Air

The following section will present descriptions of heat recovery equipment that has been investigated for this application. The initial exhaust that will be studied is from the hot water seal tanks. This has been selected because of the temperature, humidity and cleanliness of the exhaust. There are no corrosive elements in the exhaust except for water vapor. The exhaust consists of 35,250 CFM of 80°F air at 80% relative humidity. The recovered heat energy will be used to preheat

50,000 CFM of fresh outside supply air to a Grant Make-up Air Heater. This will greatly decrease the amount of heat that must be added to the supply air to maintain it at 65°F entering the plant. Rotary regenerative and heat pipe non-regenerative types of heat recovery equipment will be investigated.

Rotary Regenerative Type

A rotary air-to-air regenerative heat exchanger, sometimes called a heat wheel, is a revolving cylinder packed with air permeable media having a large surface exposed to the air stream. As the cylinder is rotated, adjacent streams of exhaust air and incoming supply air pass through it. The transfer of sensible heat is caused by the dry bulb temperature difference of adjacent air streams.

For heating, as each segment of the packed cylinder rotates through the exhaust air stream, heat is picked up. The cylinder segment then revolves through a cut-off zone, which prevents the flow of each stream into the other, and enters the incoming air stream where the heat is given up. Finally, it passes through another cut-off zone and re-enters the exhaust air stream to again pick up heat, making a continuous process.

Rotary air-to-air heat exchangers may be rated on their effectiveness to recover: (1) sensible heat (dry-bulb temperature), (2) latent heat (humidity ratio), and/or (3) total heat. Effectiveness, a term which indicates the performance of an exchanger is expressed as:

$$e = \frac{W_s(X_1 - X_2)}{W_{min}(X_1 - X_3)} \qquad (3)$$

where:

e = sensible, latent, or total heat effectiveness

X = dry-bulb temperature, humidity ratio, or total heat, respectively, at the locations indicated in Figure 7

W_s = mass flow rate of supply, pounds of dry air per hour

W_e = mass flow rate of exhaust, pounds of dry air per hour

W_{min} = minimum value of W_s or W_e

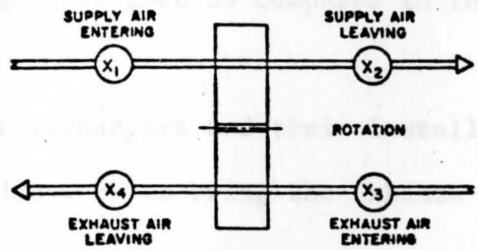


Fig. 7. Scheme of Air Flow for Rotary Air-to-Air Heat Exchanger.

The recovered or leaving supply air condition then is:

$$X_2 = X_1 - e\frac{W_{min}}{W_s}(X_1 - X_3) \tag{4}$$

The recovered or leaving exhaust air condition can be shown to be:

$$X_4 = X_3 - e\frac{W_{min}}{W_e}(X_3 - X_1) \tag{5}$$

When supply and exhaust mass flows are equal, the terms W_s/W_{min}, W_{min}/W_s, and W_{min}/W_e drop out of the above equations. The effectiveness of a rotary air-to-air heat exchanger is a function of the air mass flow, the air mass flow ratio and the energy transfer characteristics of a particular heat exchanger. Because of this, values must be established for each individual exchanger.

To make a preliminary judgment as to the feasibility of utilizing rotary regenerative air-to-air exchangers, an analysis of the overall system must be made at the following two conditions:

1. Winter design, which establishes the level of initial heating equipment saved, preheat coils not needed, etc.
2. Winter average, which establishes the level of savings in winter operating costs.

The savings must then be compared to the following additional costs incurred:

1. Cost of heat exchangers and their installation.
2. Additional ductwork to bring the exhaust and outdoor air streams adjacent.
3. Additional fan horsepower to overcome the pressure drops across the unit, and, occasionally, an additional fan.
4. Operating cost of the units. For normal ventilation applications, the units are run by fractional horsepower units.
5. In some cases, the value or cost (or both) of the floor area may become involved in the analysis, although in the majority of cases the units are located on the roof or on pads adjacent to the process where space is a negligible consideration.

For the usual ventilation system, the following points should be considered:

1. The outdoor air and exhaust air streams must be counterflow for maximum efficiency. Various configurations of equipment are available to permit horizontal air flow with either side-by-side or over-and-under air streams. Units are also available for vertical air flows.

2. Provision should be made for draining any water or snow that might accumulate by condensation or by entry from the weather.
3. During the summer, exhaust air from the space after going through the unit is still at a relatively low wet-bulb temperature and is ideal for use as air for cooling towers or evaporative condensers.
4. For system operating economy, multiple fans may be used.

Regenerative air-to-air heat exchangers are used extensively in various industrial processes. The economic feasibility of inserting a unit in an existing system depends primarily on the savings in operating cost. These savings are dependent on the required volume, temperature, and specific humidity delivered to the process; the volume, temperature, and specific humidity of the exhaust; hours of operation; cost of energy; and local design temperatures.

Condensation on the heat transfer media will occur when the dry-bulb temperature of one airstream is below the dew point of the other airstream. Under heating conditions with subfreezing outdoor air temperatures, icing or frost build-up on the media may occur if the media temperature is below 32°F and the dew point temperature of the exhaust air is above 32°F.

Under severe climatic conditions, some means of preventing frost build-up on the wheel may be required. In areas subject to frost conditions over extended periods of time, provision should be made to preheat the supply air. Under less severe conditions, an alternative method of preventing frosting is to reduce the effectiveness of recovery by either bypassing a portion of the cold supply air or by speed control. Frost control by either method may require a large reduction in the unit's effectiveness, and the heating coils must be sized for the additional load.

The main drawback of the regenerative heat wheel is cross contamination of exhaust and supply air streams. This is a major consideration if the system is handling toxic fumes. The media of the heat exchanger is largely empty. While the media is in the exhaust air stream, the void space is filled with exhaust air. During rotation, some entrained exhaust air being enclosed by partitions and air seals is carried to the supply air side. The supply air stream is in the opposite direction from the exhaust air stream. This pushes the exhaust air out of the media, where it is mixed with the supply air.

It is possible to reduce transfer of this entrained volume of air by purging from the supply to the exhaust. This is accomplished by arranging the partitions and air seals so that entrained exhaust air is purged from the heat transfer material, captured in a purge section, and returned to the exhaust air stream. Tests reveal a properly designed purge operating in a system having a negative pressure differential from the supply to exhaust duct will reduce the cross flow contamination to less than one percent by volume for randomly packed exchangers and less than 0.2 percent by volume with directionally oriented media.

Air-to-air rotary heat exchangers are packed with a coarse knitted metallic mesh, synthetic felt or an open corrugated non-hygroscopic or hygroscopic materials, so most airborne dusts pass through the heat transfer cylinder. Fine, dry, inert dusts in relatively small quantities present no problem in a counterflow arrangement.

The air flow through the rotating exchanger reverses through each revolution of the cylinder; therefore, the heat transfer material is subjected to alternating directions of air flow, providing inherent self cleaning. If the contaminants in the exhaust air are also sticky, greasy, fibrous, or are present in considerable quantities, a suitable filter should be installed ahead of the exchanger. Roughing filters for outdoor air are recommended for elimination of leaves and large particles of foreign materials. For the metallic exchanger only, steam, compressed air or hot water may be used for cleaning the wheel.

Two manufacturers of rotary heat exchangers were invited to submit bids on this application. They are the Wing Company of Linden, New Jersey and the Carnes Company of Verona, Wisconsin. The engineering staff of each company made selections that were very similar in size. They both use computer programs for application calculations.

The Wing Company quoted the Model WC 4770 Correx wheel. The wheel surface is manufactured of corrugated metal and is directional thus producing laminar flow. The total face area is 92 square feet. This will provide an actual face velocity of 766 feet per minute in the exhaust side and 1090 feet per minute in the supply side. The mass flow weighted effectiveness ($e \times W_{min}/W_s$) is equal to .527. Additional pressure drop in the supply will be .93 inches of water and in the exhaust will be .66 inches of water. See Appendix, page 58 and 59.

The Carnes Company quoted the Model T 144 MS wheel. The wheel surface is manufactured of knitted and corrugated aluminum wire media, folded and skewered into wedge sections. The media has randomly oriented air passages. The total face area is 113 square feet. This will provide an actual face velocity of 624 feet per minute in the exhaust side and 885 feet per minute in the supply side. The mass flow weighted effectiveness is equal to .540. Additional pressure drop in the exhaust will be .65 inches of water and in the supply will be 1.1 inches of water. With the randomly oriented air passages, preheat of air will be required when the outside supply air temperature is 10°F or below. If preheat is unacceptable, the wheel would not be used at outside air temperatures of 10°F or below and thus no heat energy recovery would take place during this time. See Appendix, pages 60 and 61.

Heat Pipe Nonregenerative Type

The heat pipe air-to-air recovery unit is a nonregenerative super conducting device that has no moving parts. It is very similar in appearance to a dehumidification coil with a partition separating the face into two equal sections. When installed in a system, the ducts are arranged so that one air stream flows through one side of the unit, and the other air stream flows through the opposite side. Heat is then transferred from stream to stream. Although the heat pipes span the width of the unit, a sealed partition separates the two airstreams, preventing any cross contamination between them. Energy from the hot air is transferred by the heat pipes to the other side of the exchanger where it is captured by the cold air, thereby warming it.

The unit is constructed of an array of finned tubes; each tube sealed at both ends. These tubes are the actual heat pipes. A heat pipe is a tube which has been fabricated with a capillary wick structure, evacuated, filled with a refrigerant, and permanently sealed. Thermal energy applied to either end of the pipe causes refrigerant at that end to vaporize. The refrigerant vapor then travels to the other end of the pipe where thermal energy is removed, causing the vapor to condense into liquid again, thereby giving up the latent heat of condensation. The condensed liquid then flows back to the evaporator section to be reused, thus completing the cycle. See Figure 8 for a schematic of the heat pipe cycle.

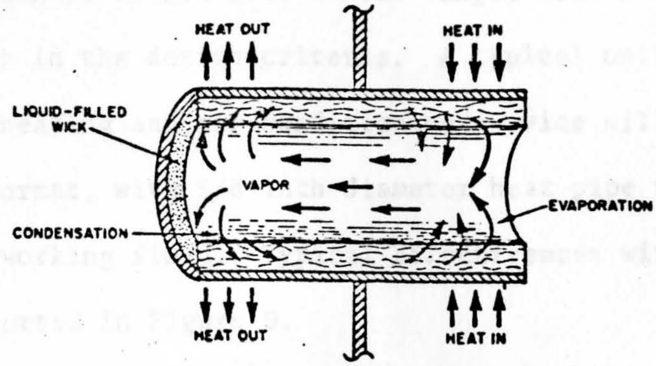


Fig. 8. Heat Pipe Schematic

The heat pipe is a completely reversible isothermal device that operates in a closed loop condensation/evaporation cycle which is continuous as long as there is a temperature gradient to drive the process.

Gravity can be utilized to assist in returning the condensate to the evaporator section by operating the heat pipe on a slope with the hot (evaporator) side below horizontal. Conversely by placing the heat pipe on a slope with the evaporator above horizontal, gravity retards the condensate flow. Changing the slope of the heat pipe thus provides a means to control the heat pipe capacity, and in turn, the performance of a heat pipe exchanger.

A heat pipe heat exchanger is basically a sensible heat transfer device, although condensation on the fins does allow latent heat transfer, resulting in improved exchanger performance. The equations for effectiveness and leaving supply air condition presented in Chapter III, page 21 can be used for calculations.

The operating effectiveness of a heat pipe recovery unit is a function of at least fifteen inter-related parameters. The interaction of any one of these with any other parameter or set of other parameters represents a trade-off point in the design of a unit. Design effectiveness can be varied over a wide range, with a fabrication cost figure implicit in the design criteria. A typical unit fairly well optimized for heating and air-conditioning service will be built on a six-row coil format, with 5/8 inch diameter heat pipe tubes and Refrigerant 12 as the working fluid. Typical effectiveness with respect to face velocity is plotted in Figure 9.

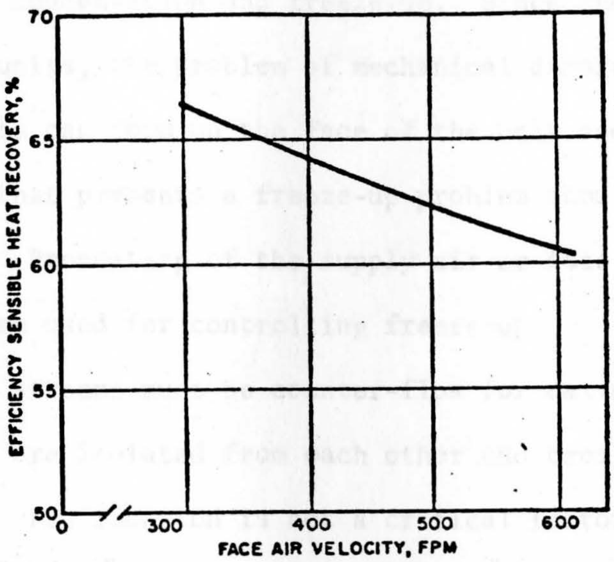


Fig. 9. Efficiency of a Heat Pipe Exchanger

Conditions similar to those for rotary equipment govern the feasibility of applying the heat pipe air-to-air recovery units. The following factors should be considered:

1. The units are a sensible heat recovery device. They should be used where latent heat recovery is not desired, or is not important.
2. The air streams are totally isolated from each other, so there can be no cross contamination. Units can be fabricated with the partition brazed or welded to the tubes for critical applications.

3. No electric power is required, and there are no rubbing surfaces, a safety consideration where combustible vapors or dusts are involved.

4. Due to the compact nature of these units, applications in addition to exhaust air heat may be considered.

Heat pipe recovery units perform like bare metallic rotary units with respect to condensation and freeze-up. Since there are no moving parts in these units, the problem of mechanical damage is minimized, but frozen condensate can form on the face of the unit and stop air flow. Each situation that presents a freeze-up problem should be evaluated on its own merits. Preheating of the supply air or face-and-bypass efficiency modulation are used for controlling freeze-up.

The air streams must be counter-flow for maximum heat transfer. The air streams are isolated from each other and cross contamination does not exist. Fan location is not a critical factor.

Heat pipe recovery units require the same filtration protection that a dehumidification coil would require when operating in the same environment. Fin pitch and the number of rows are the governing factors. Usual cleaning methods can be used (steam or pressurized hot water), however, the manufacturer should be consulted as to the maximum temperature to which the coil should be subjected.

There are two manufacturers of heat pipe recovery units. They are the Q-Dot Corporation of Dallas, Texas and Isothermics Incorporated of Augusta, New Jersey. Both companies have submitted bids on this application. They have computer optimization programs to select the best equipment for the application with maximum efficiency and least cost of equipment as the primary objectives.

The Q-Dot Corporation has quoted three 5 row/14 fin per inch type AC units to be stacked on top of each other. Each unit is 192 inches long and 40.5 inches high. Total face area for the three units is 162 square feet. The partition will be placed so that 95.25 square feet will be in the supply side and 66.75 square feet will be in the exhaust side. This will provide an actual face velocity in both the supply and exhaust of 525 feet per minute. The units will be fabricated from one inch OD aluminum tubes with corrugated aluminum fins and will be supplied with a corrosion resistant phenolic coating.

The Q-Dot installation will provide a mass flow weighted effectiveness ($e \times W_{min}/W_s$) equal to an average over the supply temperature range of .66. Additional pressure drop in the supply will be 1.00 inches of water and in the exhaust will be 1.48 inches of water. See Appendix, page 62. The following table enumerates the design conditions for this system.

TABLE 4

DESIGN CONDITIONS OF Q-DOT HEAT RECOVERY SYSTEM

Design Condition	-2°F Supply Temp.	37°F Supply Temp.
Exhaust Flow Ent.	35,250 CFM	35,250 CFM
Exhaust Temp. Ent.	80°F	80°F
Recovery Effectiveness	.6996	.6677
Supply Temp. Ent.	-2°F	37°F
Supply Flow Ent.	50,000 CFM	50,000 CFM
Enthalpy Exhaust Ent.	38.9 BTU/1b	38.9 BTU/1b
Enthalpy Supply Ent.	-.5 BTU/1b	8.9 BTU/1b
Supply Temp Leav.	55.4°F	65.7°F

TABLE 4 (CONT.)

Design Condition	-2°F Supply Temp.	37°F Supply Temp.
Exhaust Temp Leav.	47.5°F	63.4°F
Enthalpy Supply Leav.	13.3 BTU/lb	15.8 BTU/lb
Enthalpy Exhaust Leav.	18.9 BTU/lb	28.9 BTU/lb
Heat Saved	3,097,872 BTU/hr	1,550,493 BTU/hr
Moist. Cond. Out	28.5 lb/min	14.2 lb/min

The Isothermics Company has quoted three 6 row/12 fin per inch units to be stacked on top of each other. Each unit is 151 inches long and 52 inches high. Total face area for the three units is 141 square feet. The partition will be placed so that 70.5 square feet will be in the supply side and 70.5 square feet will be in the exhaust side. This will provide an actual face velocity of 500 feet per minute in the exhaust side and 709 feet per minute in the supply side. The units will be fabricated from 5/8 inch OD aluminum tubes with corrugated aluminum fins and will be supplied with a corrosion resistant phenolic coating.

The Isothermics installation will provide a mass flow weighted effectiveness (e x W_{min}/W_s) equal to an average over the supply temperature range of .65. Additional pressure drop in the supply will be 1.3 inches of water and in the exhaust will be .98 inches of water.

Installed Costs of Heat Recovery Systems

Three contractors have submitted bids for this system. All systems use basically the same ductwork design but they differ in the type of heat recovery equipment. See Appendix, page 63 for sketch of general layout of system. The bids received for the heat recovery system are as follows:

1. Thompson Heating using the Q-Dot heat pipe - $39,554.00
2. A. A. Samuels using the Carnes regenerative wheel - $32,780.00
3. A. A. Samuels using the Wing regenerative wheel - $37,345.00
4. Woodward Heating using the Isothermics heat pipe - $38,685.00
5. Woodward Heating using the Q-Dot heat pipe - $38,685.00

The heat recovery equipment and ductwork represents additional roof loading of about 6,000 pounds. Boardman Steel has submitted a bid of $1,950.00 including freight to furnish the beams, bracing and pipe posts to reinforce the roof to enable it to carry the additional loading. This reinforcing of the roof structure would be done above the roof and would not involve any interruption of production in the plant. The cost of erection and rigging of the reinforcing steel has been quoted by Diamond Steel at $1,406.00 and by the Woodward Company at $1,165.00.

The heat recovery equipment and ductwork will add static pressure drop to the exhaust fan and to the Make-up Air Heater. This will necessitate a v-belt drive change on the exhaust fan at a cost of $160.00. It will also require increasing the motor size on the Make-up Air Heater to 50 Hp (from 30 Hp) and a v-belt drive change. The existing motor starter has enough capacity for the larger motor but the branch circuit wire size must be increased to #3 gauge. Total cost of the motor, v-belt drive and electrical work is $1,700.00. An additional $600.00 is required for electrical work for the rotary regenerative type wheel.

CHAPTER IV

COMPUTER MODEL OF ENERGY RECOVERY AND EXHAUST RETURN SYSTEM

Description of Computer Program

A computer model has been developed to determine the optimum selection of the heat recovery system and to tabulate the energy usage with and without heat recovery. Average weather data from the Youngstown area was used as input data for the computer model. This is data that is based on an average over the last twenty-five years weighted to take into consideration extraordinarily extreme weather conditions.

The computer language used for the model is Fortran IV. See Appendix, page 64 for complete program. The calculations performed in the model are based on the following equations:

1. TF = TO + [E x (TE - TO)]

   TF = recovered air temperature (°F)

   TO = outside air temperature (°F)

   E = weighted effectiveness of heat exchanger

   TE = temperature of exhaust air (°F)

2. QREQ = CFM x 1.08 x (65 - TF) x HN

   QREQ = quantity of energy (BTU's) required to heat the specified quantity of air to 65°F using a heat exchanger

   CFM = quantity of air involved in particular calculation (cubic feet per minute)

   1.08 = factor based on .075 pounds per cubic foot of standard air, .24 BTU per pound F° specific heat and 60 minutes per hour

   TF = recovered air temperature (°F) from equation 1

   HN = Total number of hours for the month at specific temperature TO

3. QNOR = CFM x 1.08 x (65 - TO) x HN

QNOR = quantity of energy (BTU's) normally required to heat the specified quantity of air to 65°F

CFM = quantity of air involved in particular calculation (cubic feet per minute)

1.08 = same factor as in equation 2

TO = outside air temperature (°F)

HN = total number of hours for the month at specific temperature TO

4. QSAV = CFM x 1.08 x (65 - TO) x HN

QSAV = quantity of energy (BTU's) required to heat the specified quantity of air to 65°F that would be saved due to the recirculation system

CFM = quantity of air involved in particular calculation (cubic feet per minute)

1.08 = same factor as in equation 2

TO = outside air temperature (°F)

HN = total number of hours for the month at specific temperature TO

The program printout shows in tabulated form the total value of QREQ, QNOR, and QSAV for each month of the winter heating season. The values of the effectiveness of the four heat exchangers were used to compare the energy savings that would be produced by the different systems. With the Carnes Wheel, frosting would occur at outside air temperature of 10°F or less. It was decided to stop the wheel at this point and not risk damaging the wheel or fan. Electric preheat was not acceptable because with the present substation, not enough service was available. Also, because of the purge requirement (3% of supply volume) with the rotary regenerative type, QSAV was based on a lesser volume than with the heat pipe nonregenerative type. This was done since the Make-up Air Heaters without recovery could not have volumes reduced quite equal to the volume of the recirculation system and still maintain the required air balance.

Simulation Results

The energy saved for each heat recovery system model will be equal to the difference in energy required to heat the specified volume of air without recovery (QNOR) less the energy required with recovery (QREQ). The total energy saved for the combined recovery and recirculation systems will be this difference (QNOR - QREQ) plus the energy saved due to the recirculation system (QSAV).

The following table presents the results of the computer simulation for the heating season of October through April.

TABLE 5

COMPUTER SIMULATION RESULTS

Manufacturer	Effectiveness	QNOR-QREQ (10^9 BTU)	QSAV (10^9 BTU)	Total Q Saved
Wing-wheel	.527	5.4075	2.8527	8.2598
Carnes-wheel	.540	5.1813	2.8523	8.0336
Isothermics-heat pipe	.650	6.7703	3.0738	9.8441
Q-Dot-heat pipe	.660	6.8745	3.0738	9.9483

The East Ohio Gas Company has allocated specified quantities of natural gas by monthly allocations. The allocations differ from month to month and are determined through mutual agreement at the beginning of the fiscal gas year in March. The allocation can be exceeded by 10% for one month as long as the total allocation for that quarter does not exceed the quarter's allocation. At the present time, the natural gas allocation schedule at General Extrusions requires the use of one truckload of propane (equal to 7.2×10^8 BTU at 100% combustion efficiency) during every month. The natural gas allocation schedule for 1976 will be changed to reflect the energy conservation program. Thus the first 8.64×10^9 BTU saved during the year will be costed at current propane prices (\$3.80 per 10^6 BTU) and the balance will be costed at current natural gas prices (\$1.26 per 10^6 BTU). These costs are based on 100% combustion efficiency. From the descriptive literature of the W.C. Grant Make-up Air Heaters and the operation experience, it was determined that an efficiency of 95% would be used for the energy dollar savings calculations. Thus at 95% combustion efficiency propane costs \$4.00 per 10^6 BTU and natural gas costs \$1.33 per 10^6 BTU. Table 6 presents the energy dollar savings from each system.

TABLE 6

ENERGY DOLLAR SAVINGS

Manufacturer	Total Q Saved (10^9 BTU)	Total Dollars Saved	Total Systems Cost
Wing-wheel	8.2598	\$33,039.00	\$47,344.00
Carnes-wheel	8.0336	\$32,134.00	\$42,779.00
Isothermics-heat pipe	9.8441	\$36,161.00	\$47,843.00
Q-Dot-heat pipe	9.9483	\$36,396.00	\$47,843.00

CHAPTER V

ECONOMIC ANALYSIS

Life Cycle Costing

The concept of life-cycle (long term) costing is becoming increasingly important to industry. In life-cycle costing, instead of considering only the initial (capital) cost of a project, you include all the costs of owning a project over its lifetime. These costs fall into two general categories:

1. Initial, or capital, cost
2. Operating and Maintenance (O and M) cost.

Category 1. includes all the front end project costs - everything from land and construction costs to engineering and legal fees. Category 2. can include, in addition to annual operating and maintenance costs, such items as real estate taxes and insurance, if these costs appear as variables that could affect the owner's economic choice among different alternatives.

The concept of life-cycle costing involves the total cost of owning a building or piece of equipment, throughout its assumed useful life. It differs from the traditional concept, expressed in conventional lump-sum bidding, of focusing exclusively on first cost as the sole economic criterion for comparing construction alternatives. For accurate comparison of alternatives, you must reduce costs to a common base. Normally, this is the total annual cost comprising (a) amortization for the capital investment, plus (b) operating and maintenance (O and M) cost, plus such potential annual variables as insurance premiums or real estate taxes, if they differ for the compared alternatives.

Benefit - Cost Analysis

A benefit - cost analysis can provide a rational basis for choosing among alternatives. It can be utilized to determine which of two or more alternative systems provides the best cost benefit.

When using benefit - cost analysis to select between two systems, a simple formula can be used to determine if the additional costs of the more expensive system are merited in light of long term cost factors. The result is a benefit - cost ratio which, if it exceeds 1, indicates that extra initial expenses will result in long term savings.

The systems that will be compared based on this benefit - cost ratio will be the Wing wheel and Q-Dot heat pipe comparison and the Carnes wheel and Q-Dot heat pipe comparison. The Q-Dot heat pipe is obviously the better selection over the Isothermics heat pipe since the installed and maintenance costs are the same and the energy savings is greater with the Q-Dot system. A 20 year useful life and a 10% interest rate is used to calculate the capital recovery factor (a factor that multiplied by the total loan amount yields the annual payment necessary to repay debt). In this case, it will be 0.11746. Current energy costs are used throughout the 20 year life.

The following is the comparison of the Wing and Q-Dot systems:

Total First Cost Wing System $47,344

Total First Cost Q-Dot System $47,843

Q-Dot exceeds Wing by $499

Wing annual operating, maintenance and energy cost exceeds Q-Dot by $3,357

Amortization cost for additional capital investment of the Q-Dot system would be .11746 x $499 or $58.62. Therefore the benefit - cost ratio for the Q-Dot system would be:

$$\frac{\text{annual savings} \quad \$3,357.00}{\text{amort. cost} \quad \$58.62} = 57.3$$

Because the benefit - cost ratio far exceeds 1, the Q-Dot system although initially more expensive, will provide more long term savings.

The following is the comparison of the Carnes and Q-Dot systems:

Total First Cost CArnes System $42,779

Total First Cost Q-Dot System $47,843

Q-Dot exceeds Carnes by $5,064

Carnes annual operating, maintenance and energy cost exceeds Q-Dot by $4,262

Amortization cost for additional capital investment of the Q-Dot system would be .11746 x $5,064 or $594.82. Therefore the benefit - cost ratio for the Q-Dot system would be:

$$\frac{\text{annual savings} \quad \$4,262.00}{\text{amort. cost} \quad \$594.82} = 7.2$$

Because the benefit - cost ratio exceeds 1, the Q-Dot system again, although initially more expensive, will provide more long term savings.

Payback Period Analysis

From the benefit - cost analysis it is obvious that the Q-Dot system is the best selection. When making a payback period analysis, cost of debt service must be considered. The following formula is used to calculate the payback period:

$$n = \frac{\log \dfrac{s/rC}{s/rC - 1}}{\log\,(1 + r)} \tag{6}$$

where:

C = Capital cost

s = Annual operating and maintenance savings

r = interest rate

n = number of years to achieve payback

For the Q-Dot system selected, the payback period was calculated using C = \$47,843, s = \$36,396 and r = .10. The payback period (n) = 1.48 years.

Selection of Optimized System

A meeting was held with the management of the company to discuss the project. The energy savings, capital cost and payback period were reviewed. The system will take advantage of the present fixed cost of heat recovery equipment and the future increasing cost of energy. The company decided to go ahead with the project and the contract was awarded to the Woodward Company.

CHAPTER VI

CONCLUSION

Description of the Q-Dot System

The energy savings that will result from the recirculation and Q-Dot heat recovery systems will be approximately 10 million cubic feet of natural gas or propane equivalent. Energy savings will vary yearly due to varying weather conditions. Energy dollar savings will be approximately \$36,400. Based on the quoted installed cost the payback period will be 1.48 years.

The heat recovery system will require no change in plant operation since the exhaust fans are continuously operating twenty-four hours per day and seven days per week. The system will recover heat continuously if the Make-up Air Heater fan is operating. To fully utilize this recovery system this Make-up Air Heater should operate continuously. Only one of the Make-up heaters operates on weekends, so it will be the one with the heat recovery system.

This is the largest single installation of Q-Dot equipment in the country. This system handles a total of 85,000 CFM, exhaust and supply. The system is also the only one in the country handling such a low temperature but high humidity exhaust and because of this has received considerable engineering study by Q-Dot personnel to compare theoretical and actual values of the system's operation.

As a by product of the energy recovery system, a significant quantity of water vapor will condense on the heat recovery coil. The system will produce about one hundred and fifty gallons of water per hour for average winter outside air temperature of 35°F. As the outside air temperature decreases, more water will condense. This water can be recycled back into the tanks. The piping system will be completed by the Woodward Company.

Installation

The contract was awarded to the Woodward Company on November 25, 1975. Actual construction of the recirculation system and the ductwork required for the recovery system began on December 10, 1975. The recirculation system and the recovery ductwork was completed on December 23, 1975 with no problems during installation.

Installation work stopped because of delay in manufacture of Q-Dot heat pipes. The Q-Dot units were received on January 12, 1976 and the placement of these units on the roof was completed that day. Work continued on the system and all transition ductwork was completed on January 23, 1976. The system was fully operational on January 26, 1976. During the installation the operation of the plant was not affected. The exhaust fan was not operating for only a period of three separate eight hour days. The installation of the larger motor on the Make-up Air Heater required only one eight hour day.

The total installation proceeded with no major problems. On February 2nd, final air balancing of the system was to be completed. General Extrusions Inc.'s local union contract ended on February 1st and the union struck the plant. The system was shut down for the duration of the labor disruption and as of this writing final air balance and performance testing have not been completed.

Preliminary Testing of Performance

During the last day of operation of the system before the strike, testing of the effectiveness of the heat exchanger was done using Honeywell duct thermostats. These thermostats only read to an accuracy of $\pm 2°F$. But with this rough test the effectiveness of the unit was .67.

Sensitivity Analysis

A preliminary investigation of additional heat recovery equipment will be presented in this section. Additional energy saved will be costed at natural gas prices since propane is no longer required due to the heat recovery system already installed. This will provide longer payback periods. Also, because of the exhaust fans and Make-up Air units, recovery systems will have to be installed for large air volumes.

As an example, the exhaust fan-separator Mark A (see Figure 1, page 3), exhausts a relatively clean air that would not be too corrosive to the Q-Dot heat pipe. Because of the scrubber design and the equipment layout, the estimated cost to add heat recovery to Mark A and Mark E, the Make-up Air Heater, would be $65,000. Energy savings would be approximately 9.68×10^9 BTU based on a reduced mass flow weighted effectiveness of .56 because of the lower humidity and condensation conditions. Based on the cost of natural gas of $1.33 per 10^6 BTU, this would result in an energy savings of $12,874.00. Using the payback period analysis in Chapter V, the number of years to achieve payback would be 7.4. At present energy costs this payback period would not be acceptable because the capital could be invested in other projects that would pay a better return.

This project could become feasible if natural gas costs increased and/or interest costs decreased. Based on natural gas costs doubled and interest at 8%, the payback period would be 2.93 years. Based on natural gas costs tripled and interest at 8%, the payback period would be 1.87 years.

As can be seen from the above estimate, at present energy costs it is not feasible to install additional heat recovery equipment.

This thesis has shown that energy conservation techniques can be economically applied in industrial situations and can provide an answer to the national problem of energy shortages.

APPENDIX

ANODIZING

1. Definition: Anodizing is an electrochemical reaction that builds a hard transparent oxide film on the surface of the aluminum. By making the aluminum part an anode in electrolyte solution, a dense anodic coating is formed. The acid electrolyte provides oxygen ions that react with the aluminum ions to form the oxide coating. The result is a coating that starts on the outside of the metal and grows inward. This coating is clear, transparent, and colorless, with about the hardness of a diamond.
2. Purpose: To make the metal corrosion resistant, wear resistant, and give extra hardness. Because the anodize coating is porous, the metal can be dyed any number of colors for an attractive finish.
3. Process:
   a. Rack Parts
      1. To maintain a good electrical contact during anodizing, firm clamping is necessary. (Clamping or pin marks will be noticeable after part is anodized. They will be more pronounced on longer and heavier material.)

   b. Clean Parts

   c. Etch, brite dip, and anodize
      1. Any combination of these three can be used, but they must remain in above order.

   d. Dye (color)
      1. The anodize coating is porous (500 billion pores per square inch) and will allow dye to permeate metal.
      2. Virtually any color is possible. Popular colors now are gold, bronze, pewter, black, blue and red.

3. All dyes will fade to some extent with the exception of the GE 300 series, which is an inorganic dye.

e. Color Seal

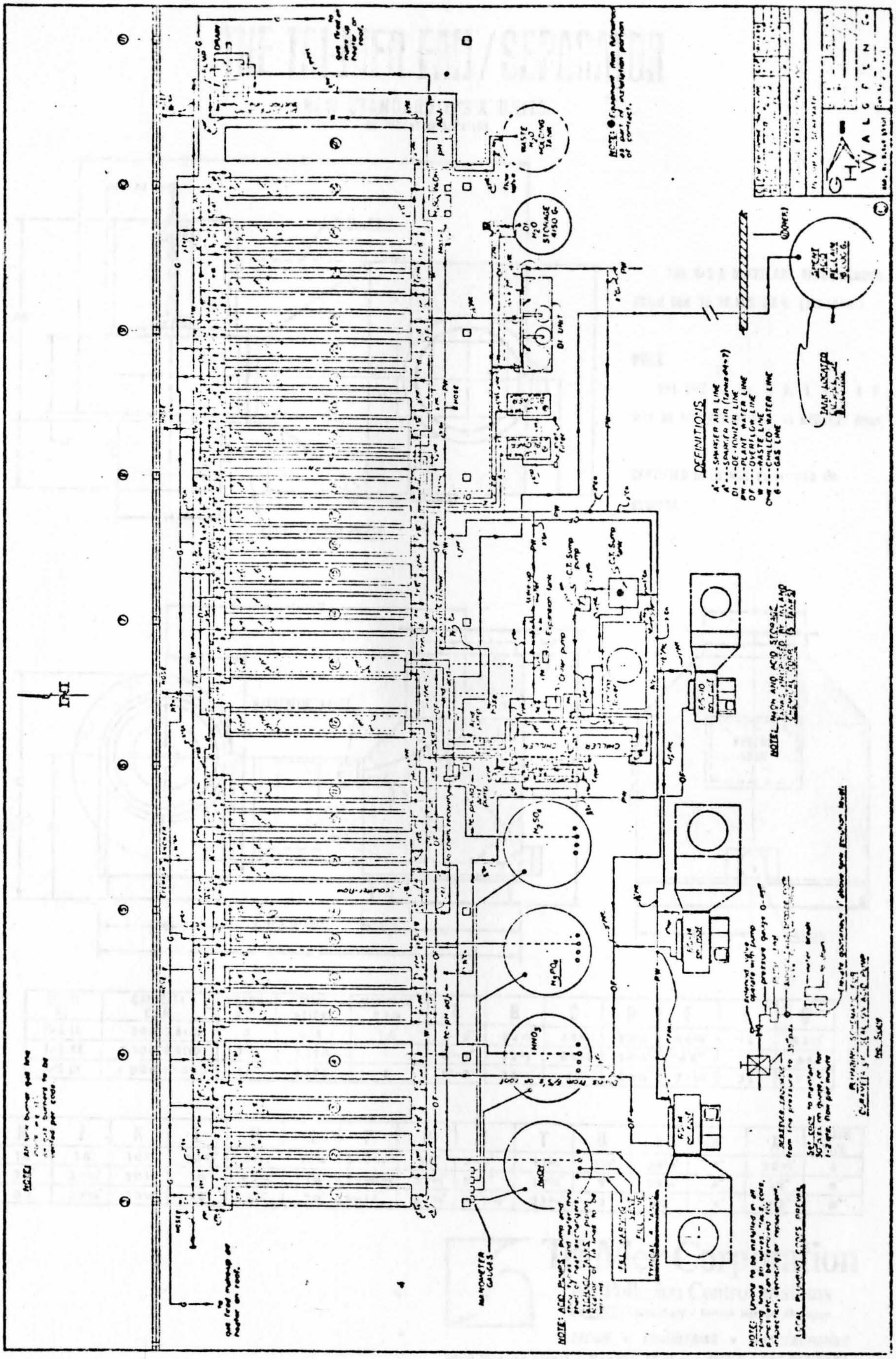
1. Sets the dye

f. Hot water seal

1. Seals the metal

2. Sealing time increases in conjunction with anodizing time.

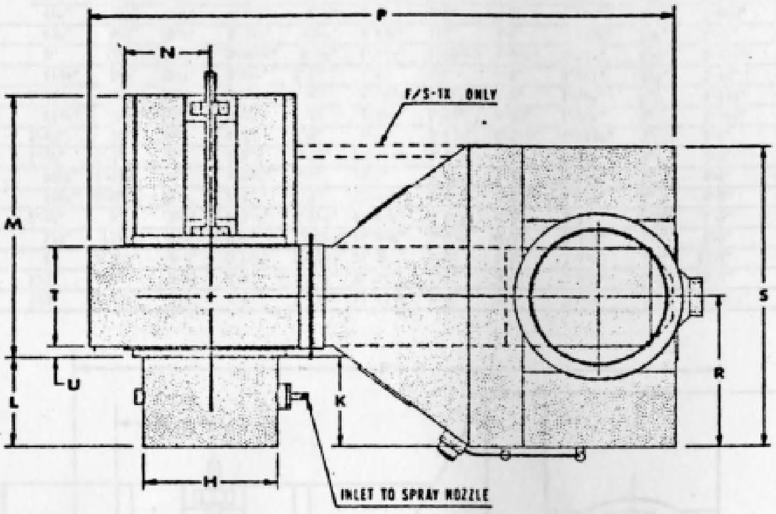
g. Parts are dried and unracked.



THE TRI-MER FAN/SEPARATOR

NEW STANDARD F/S-X UNITS

U.S. PATENT Nº 3.611.604

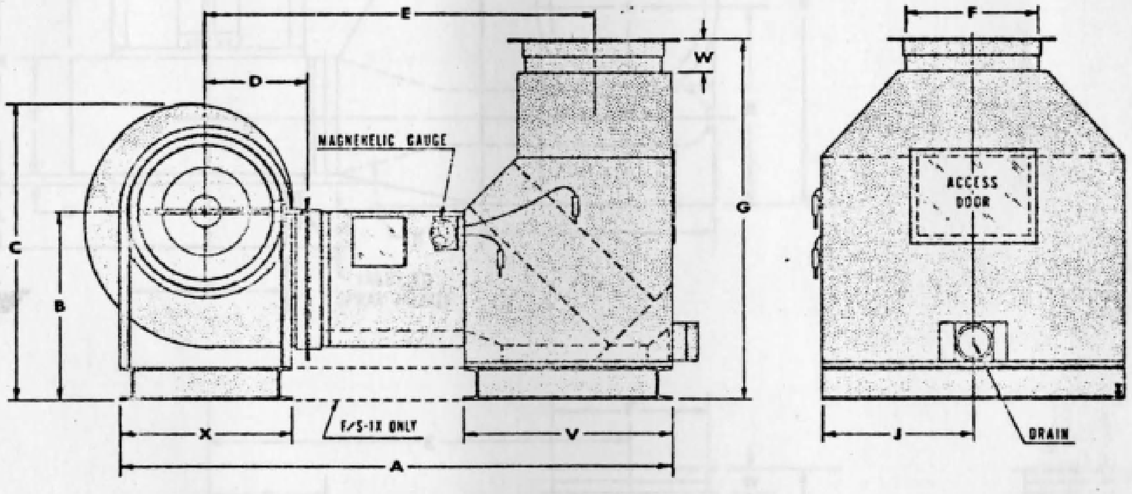


THE F/S-X UNITS ARE RECOMMENDED FROM 500 TO 10.300 C.F.M. (SEE CHART)

NOTE:

THE F/S-¼, ½, 1, 2, 3, 4, 5 & 6 MAY BE ORDERED BY SPECIAL REQUEST ONLY.

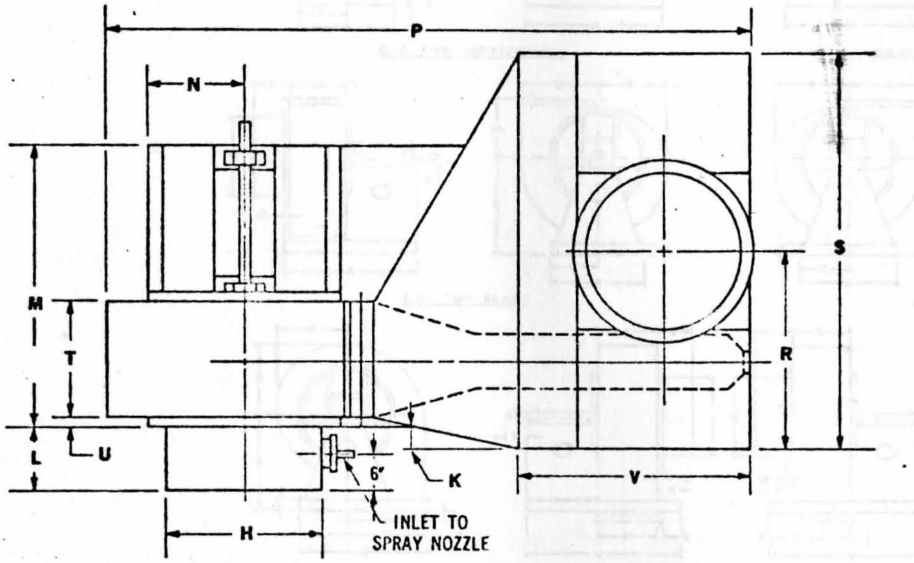
CERTIFIED DRAWINGS FURNISHED ON REQUEST.



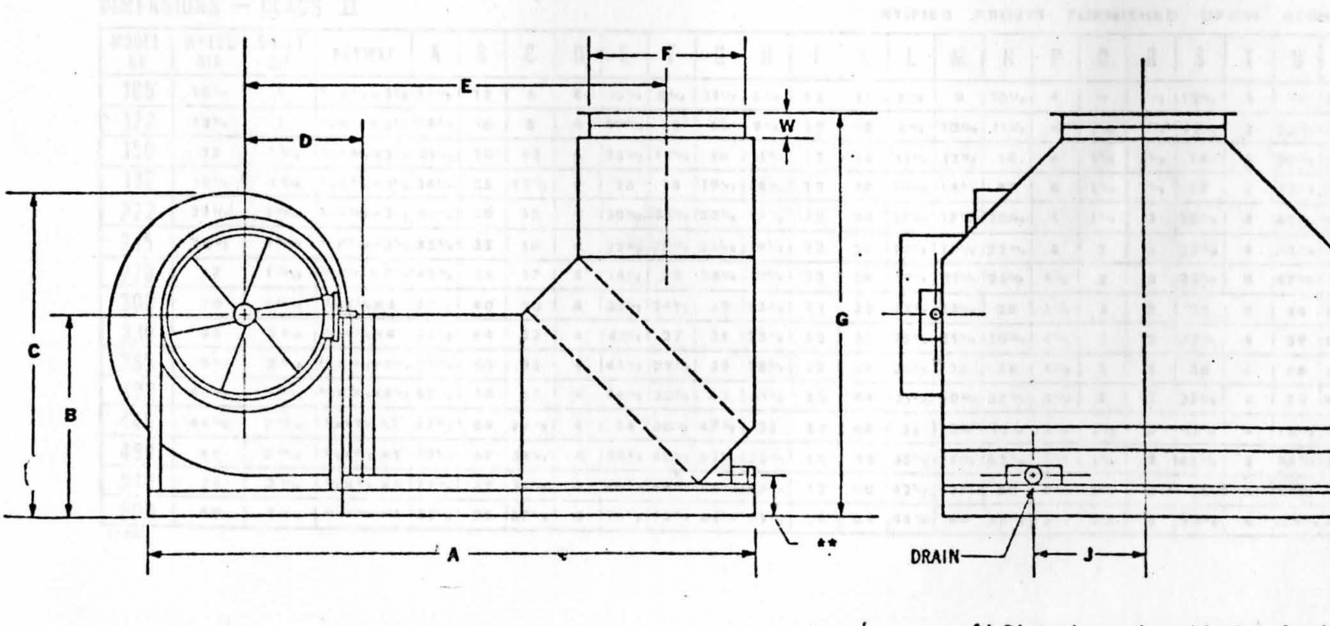
MODEL Nº	CAPACITY C.F.M.	MOTOR H.P.	TOTAL WEIGHT	MAXIMUM G.P.M.	A	B	C	D	E	F	G
F/S-1X	500-2.500	3	615	1.5	5'-5"	22½"	34¾"	12¼"	46¼"	16"	43½"
F/S-4X	2.300-6.900	7½	1,190	4	8'-4"	32¾"	52¾"	19¼"	69"	26"	64"
F/S-6X	6.000-10.300	15	1,875	6	9'-3"	38½"	63"	23½"	71¼"	32"	74"

H	J	K	L	M	N	P	R	S	T	U	V	W	X	DRAIN SIZE
16"	18"	10½"	10¾"	31½"	10¼"	5'-8¾"	18"	3'-0"	11¾"	1¼"	24½"	4"	20½"	3"
26"	32½"	20½"	10"	45¼"	16"	8'-10¼"	32½"	5'-5"	19¼"	2"	38¾"	4"	32"	6"
32"	37¼"	23½"	10"	51"	20"	9'-11"	37¼"	6'-2½"	23¾"	2"	42"	4"	40"	6"

H	J	K	L	M	N	P	R	S	T	U	V	W	Maximum GPM Required	Drain Size	Total Weight
11"	3⅜"	5½"	10¾"	2'-4¼"	6"	4'-5½"	1'-2⅛"	2'-4½"	8½"	1¼"	2'-¾"	3"	½	2"	360
1'-1"	4¾"	7¾"	10¾"	2'-4½"	8¼"	4'-6"	1'-6"	3'-0"	9⅝"	1¼"	1'-9½"	3"	½	2½"	425
1'-4"	6⅜"	4½"	10¾"	2'-7½"	10¼"	5'-8¾"	1'-6"	3'-0"	11¾"	1¼"	2'-1½"	4"	1½	3"	615
1'-7"	9"	5"	10½"	2'-10¼"	1'-½"	6'-6½"	1'-10¾"	3'-9½"	1'-2¼"	1½"	2'-4½"	4"	2	4"	770
2'-0"	11⅜"	2½"	10½"	3'-5¼"	1'-3"	7'-8"	2'-0"	4'-0"	1'-5¼"	1½"	2'-10"	4"	3	4"	1,025
2'-2"	1'-6"	3"	10"	3'-9¼"	1'-4"	8'-10¾"	2'-8½"	5'-5"	1'-7¼"	2"	3'-2¾"	4"	4	6"	1,190
2'-4"	11¼"	10⅝"	10"	4'-¾"	1'-5"	9'-3¼"	2'-11"	5'-10"	1'-9¼"	2"	3'-1½"	4"	5	6"	1,605
2'-8"	1'-½"	10"	10"	4'-3"	1'-8"	9'-11"	3'-1¼"	6'-2½"	1'-11⅜"	2"	3'-6"	4"	6	6"	1,875
2'-11"	1'-½"	1'-1"	10"	4'-7"	1'-10"	10'-7¼"	3'-4½"	6'-9"	2'-1⅞"	2"	3'-9"	4"	8	6"	2,068
3'-3"	10½"	1'-1¾"	10"	4'-9½"	2'-1"	12'-2"	3'-4½"	6'-9"	2'-4½"	2"	4'-4"	4"	9	6"	2,380
3'-8"	10¾"	1'-0"	10"	5'-2"	2'-3"	13'-½"	3'-4½"	6'-9"	2'-7⅝"	2"	4'-8"	4"	12	6"	2,930
4'-0"	1'-½"	8"	9½"	5'-7½"	2'-5½"	16'-5¼"	3'-4½"	6'-9"	2'-11"	2½"	7'-6"	4"	14	6"	3,470
4'-5"	7½"	11½"	9½"	6'-0"	2'-10½"	17'-6¾"	3'-4½"	6'-9"	3'-2½"	2½"	8'-0"	4"	17	6"	4,120
5'-0"	6"	1'-5½"	9½"	6'-5¾"	2'-10½"	21'-3½"	4'-0"	8'-0"	3'-7¾"	2½"	10'-6"	4"	20	6"	4,630
5'-4"	1'-3"	1'-2"	9½"	6'-8"	3'-1½"	21'-11"	4'-6"	9'-0"	3'-9¾"	2½"	10'-7"	4"	24	6"	5,100
5'-4"	1'-3"	1'-4½"	9½"	6'-8"	3'-1½"	22'-11½"	4'-6"	9'-0"	3'-9¾"	2½"	11'-8½"	4"	30	6"	6,600

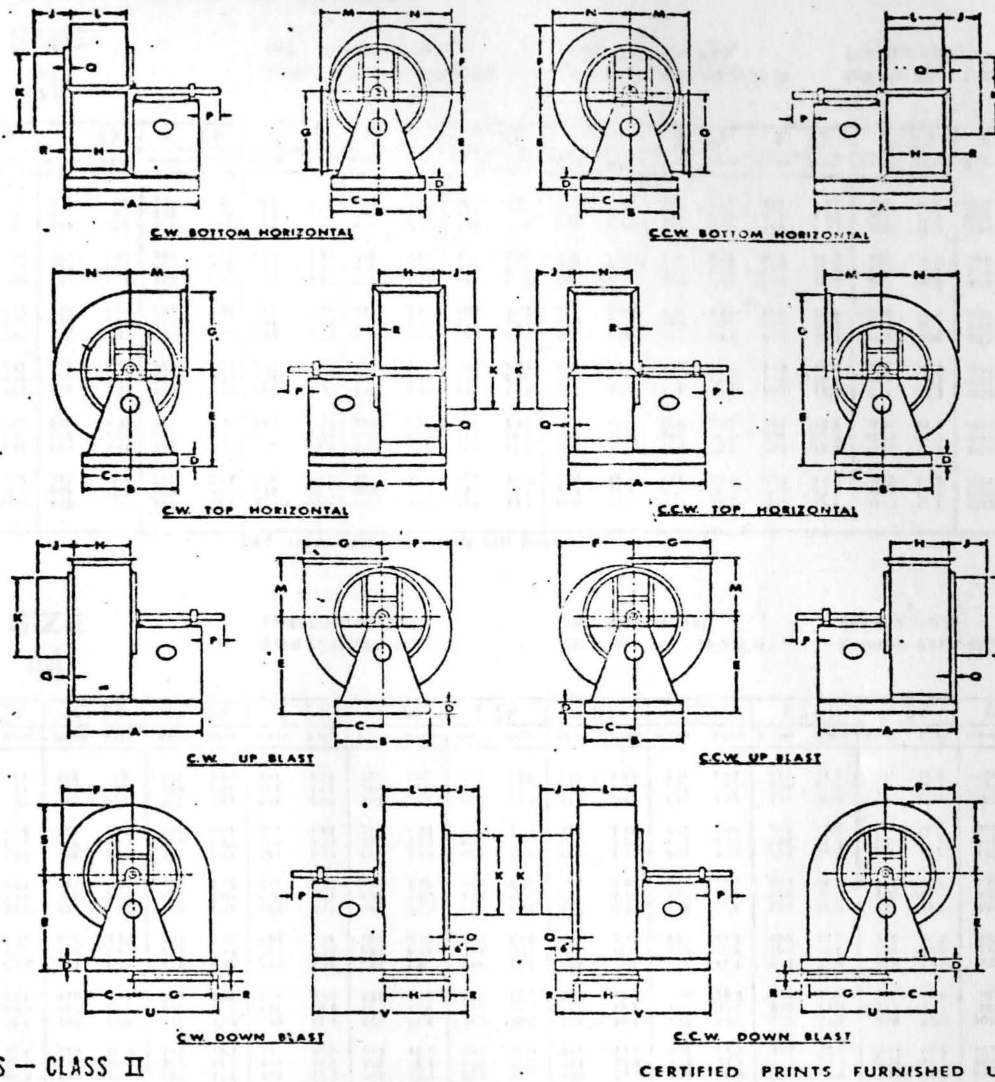


Above dimensions are not for construction. Request certified prints.



** Dimension varies with size of unit.

SWSI ARRANGEMENT 1



DIMENSIONS — CLASS II

CERTIFIED PRINTS FURNISHED UPON RE

MODEL Nº	WHEEL DIA	SHAFT DIA	KEYWAY	A	B	C	D	E	F	G	H	J	K	L	M	N	P	Q	R	S	T	U
105	10 1/2	1	1/4 × 1/8 × 3 1/2	21 3/8	12	6	4	15 1/2	8 1/4	11 1/2	8 3/8	12	11	8 3/8	9	10 1/4	4	–	1 1/2	10 1/4	3	19
122	12 1/4	1	1/4 × 1/8 × 3 1/2	28 3/8	16	8	4	19 9/16	10	13	9 5/8	12	13	9 5/8	10 1/2	11 3/8	4	1 1/4	1 1/2	11 3/8	3	22[illegible]
150	15	1 3/16	1/4 × 3/16 × 3 1/2	31 1/2	20	10	4	22 1/2	12 1/4	16	11 3/4	12	16	11 3/4	12 3/8	14	4	1 1/4	1 1/2	14	3	27[illegible]
182	18 1/4	1 7/16	3/8 × 3/16 × 3 1/2	34 1/4	25	12 1/2	4	26	15	19 1/2	14 1/4	12	19	14 1/4	14 1/2	17	4	1 1/2	1 1/2	17	3	33[illegible]
222	22 1/4	1 11/16	3/8 × 3/16 × 3 1/2	41 1/4	30	15	4	30 1/4	18 1/4	23 3/4	17 1/4	12	24	17 1/4	17 1/4	20 3/4	4	1 1/2	2	20 3/4	4	40[illegible]
245	24 1/2	1 15/16	1/2 × 3/16 × 3 1/2	45 1/8	32	16	4	32 1/8	20 1/8	26 1/8	19 1/2	12	26	19 1/2	19 1/4	22 3/4	4	2	2	22 3/4	4	44[illegible]
270	27	1 15/16	1/2 × 1/4 × 3 1/2	48 3/4	34	17	4	34 1/8	22	28 1/4	21 1/4	12	28	21 1/4	21 1/2	25 1/8	4 1/4	2	2	25 1/8	4	47[illegible]
300	30	1 15/16	1/2 × 1/4 × 4	50 7/8	40	20	4	38 1/2	24 3/8	32	23 3/4	12	32	23 3/8	23 3/8	28	5 1/4	2	2	28	4	54
330	33	2 3/16	1/2 × 1/4 × 4	54 7/8	44	22	4	41 1/2	27	35	25 7/8	12	35	25 7/8	25 5/8	30 3/4	4 1/2	2	2	30 3/4	4	59
365	36 1/2	2 7/16	5/8 × 5/16 × 4 1/2	57 1/2	50	25	4	45 1/2	29 7/8	39	28 1/2	12	39	28 1/2	28	34	4 1/2	2	2	34	4	66
402	40 1/4	2 11/16	5/8 × 5/16 × 4 1/2	62 1/8	54	27	4	49 1/2	32 7/8	43	31 1/4	12	44	31 1/2	30 5/8	37 1/2	5 1/2	2	2	37 1/2	4	72
445	44 1/2	2 15/16	3/4 × 5/16 × 5	67 1/2	59	29 1/2	4	54	36 1/4	47 3/8	35	12	48	35	34	41 1/4	5 1/2	2 1/2	2	41 1/4	4	78[illegible]
490	49	2 15/16	3/4 × 3/8 × 5	72 1/2	69	34 1/2	4	53 3/4	40 1/8	52 1/4	38 1/2	12	53	35 1/2	37 1/8	45 3/4	5 1/2	2 1/2	2	45 3/4	4	88[illegible]
540	54	3 3/16	3/4 × 3/8 × 5	77 3/8	69	34 1/2	6	66 7/8	44 1/2	58 3/8	43 3/4	12	60	43 3/4	43 1/2	50	5 1/2	2 1/2	2	50	4	94[illegible]
600	60	3 7/16	7/8 × 3/8 × 5	80 1/8	75	37 1/2	6	73 1/4	46 3/4	64 3/4	45 3/4	12	64	45 3/4	46	53 1/2	6 1/2	2 1/2	2	53 1/2	4	104[illegible]

1974

CAPACITY TABLES

SIZE 40

Wheel Diameter = 40 ¼"
Wheel Circumference = 10.54'

Inlet Diameter = 44"
Fan Outlet Area = 9.35 sq. ft.

Safe RPM = 1260
Maximum BHP = 24.6 $\left(\frac{RPM}{1000}\right)^3$

CFM	OV	¼" SP		⅜" SP		½" SP		⅝" SP		¾" SP		1" SP		2" SP		3" SP		4" SP		5" SP		6" SP	
		RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
7455	800	273	.49	300	.66	324	.83	347	1.01	370	1.21	414	1.61	571	3.62	704	6.11	817	8.93	917	12.2	1007	15.4
8375	900	295	.61	319	.81	348	.98	363	1.12	384	1.37	424	1.81	572	3.85	700	6.39	814	9.32	913	12.5	1001	15.9
9310	1000	312	.74	340	.95	361	1.15	381	1.36	400	1.57	438	2.02	575	4.12	699	6.71	810	9.73	909	13.2	1000	16.5
10241	1100	339	.89	361	1.12	381	1.35	400	1.57	418	1.81	453	2.27	582	4.44	700	7.06	808	10.2	906	13.4	997	17.2
11172	1200	362	1.06	384	1.32	402	1.56	420	1.81	438	2.06	470	2.56	592	4.79	704	7.47	807	10.5	904	14.2	993	17.7
12103	1300	386	1.26	406	1.54	424	1.81	441	2.07	457	2.34	489	2.88	603	5.21	709	7.92	809	11.2	903	14.5	990	18.3
13034	1400	410	1.45	429	1.79	446	2.08	462	2.37	478	2.65	508	3.23	617	5.66	718	8.43	813	11.6	904	15.2	959	18.9
13965	1500	435	1.75	452	2.07	468	2.37	484	2.69	499	2.99	528	3.61	632	6.15	728	9.01	820	12.2	907	15.7	989	19.6
14869	1600	460	2.04	477	2.39	492	2.71	507	3.01	520	3.35	549	4.01	648	6.66	742	9.63	828	12.9	912	16.4	992	20.3
16758	1800	510	2.72	525	3.11	539	3.47	552	3.83	565	4.18	591	4.93	685	7.91	769	11.2	848	14.4	926	18.2	1002	22.2
18700	2000	560	3.55	575	3.99	587	4.39	600	4.79	612	5.19	635	6.01	722	9.25	802	12.7	876	16.2	947	20.2	1017	24.2
20452	2200	612	4.57	626	5.07	637	5.49	648	5.91	655	6.34	680	7.22	763	10.8	838	14.5	907	18.2	974	22.3	1038	26.4
22344	2400	664	5.77	676	6.31	687	6.76	697	7.25	708	7.71	728	8.68	804	12.5	876	16.5	943	20.5	1004	24.7	1065	29.2
24206	2600	716	7.18	728	7.76	738	8.28	747	8.78	756	9.27	776	10.3	848	14.5	915	18.7	978	23.3	1037	27.4	1096	32.2
26068	2800	769	8.81	780	9.46	788	9.97	797	10.5	806	11.1	825	12.2	892	16.6	956	21.2	1016	25.8	1074	30.5	1128	35.3
27930	3000	821	10.6	832	11.4	840	11.9	848	12.5	857	13.1	873	14.3	938	19.1	998	23.9	1056	28.8	1112	33.8	1164	38.8
29792	3200	875	12.8	885	13.6	893	14.2	899	14.7	908	15.4	923	16.6	983	21.6	1042	26.8	1096	32.3	1150	37.4	1202	42.7
31654	3400	927	15.2	938	16.1	944	16.6	951	17.3	959	18.1	973	19.3	1031	24.6	1086	30.2	1139	35.6	1190	41.2	1240	46.9

BHP shown does not include belt drive loss.

SIZE 44

Wheel Diameter = 44½"
Wheel Circumference = 11.6'

Inlet Diameter = 48"
Fan Outlet Area = 11.5 sq. ft.

Safe RPM = 1150
Maximum BHP = 41.2 $\left(\frac{RPM}{1000}\right)^3$

CFM	OV	¼" SP		⅜" SP		½" SP		⅝" SP		¾" SP		1" SP		2 SP		3" SP		4 SP		5" SP		6" SP	
		RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
9112	800	246	.61	270	.82	292	1.07	313	1.24	333	1.47	373	1.98	515	4.45	635	7.52	737	11.1	827	14.8	908	18.9
10250	900	265	.75	288	.98	308	1.21	328	1.45	346	1.69	382	2.21	516	4.74	632	7.86	734	11.5	824	15.4	904	19.6
11390	1000	286	.91	306	1.17	325	1.42	344	1.68	351	1.94	395	2.49	519	5.07	630	8.25	731	11.9	820	15.9	902	20.3
12529	1100	306	1.09	326	1.38	344	1.66	361	1.94	377	2.23	408	2.81	525	5.46	631	8.69	728	12.4	817	16.5	899	21.1
13668	1200	327	1.31	346	1.63	362	1.92	379	2.23	394	2.53	424	3.15	533	5.91	635	9.19	728	12.9	815	17.2	896	21.8
14807	1300	348	1.55	366	1.91	382	2.22	397	2.55	412	2.88	441	3.54	544	6.39	640	9.74	730	13.6	814	17.8	893	22.5
15945	1400	370	1.83	387	2.21	402	2.56	417	2.91	431	3.26	458	3.97	557	6.97	647	10.4	733	14.2	815	18.5	892	23.2
17055	1500	392	2.16	408	2.54	422	2.91	437	3.31	450	3.67	476	4.43	570	7.55	657	11.1	739	15.1	818	19.4	892	24.1
18234	1600	414	2.51	430	2.94	443	3.34	457	3.73	469	4.13	495	4.94	585	8.22	668	11.8	747	15.8	821	20.2	894	25.1
20502	1800	460	3.35	473	3.82	486	4.27	498	4.71	509	5.15	533	6.06	617	9.73	694	13.6	765	17.7	835	22.2	903	27.2
22780	2000	505	4.37	518	4.91	529	5.41	541	5.89	551	6.38	573	7.38	651	11.4	723	15.6	790	19.9	854	24.6	917	29.6
25058	2200	552	5.62	564	6.24	574	6.75	584	7.26	594	7.81	614	8.88	688	13.3	755	17.8	818	22.4	878	27.3	936	32.4
27336	2400	599	7.11	610	7.75	619	8.32	629	8.91	638	9.49	656	10.7	725	15.4	790	20.3	849	25.3	905	30.4	961	35.7
29614	2600	646	8.83	656	9.54	665	10.2	674	10.8	682	11.4	699	12.7	764	17.8	825	23.1	882	28.4	935	33.7	984	39.6
31891	2800	693	10.8	703	11.6	711	12.3	719	12.9	727	13.5	743	15.1	804	20.5	862	26.1	916	31.7	968	37.5	1017	43.3
34170	3000	749	13.1	750	14.1	757	14.7	765	15.4	773	16.1	787	17.5	845	23.4	900	29.4	952	35.4	1002	41.6	1049	47.7
36448	3200	789	15.8	798	16.7	805	17.5	811	18.1	818	18.9	832	20.4	887	26.6	939	33.1	989	39.3	1036	46.1	1083	52.6
38726	3400	836	18.6	845	19.8	851	20.5	856	21.3	865	22.1	878	23.7	929	30.3	979	36.9	1027	43.8	1073	50.7	1118	57.6

BHP shown does not include belt drive loss.

SIZE 49

Wheel Diameter = 49"
Wheel Circumference = 12.8'

Inlet Diameter = 53"
Fan Outlet Area = 13.8 sq. ft.

Safe RPM = 1050
Maximum BPH = 66.8 $\left(\frac{RPM}{1000}\right)^3$

CFM	OV	¼" SP		⅜" SP		½" SP		⅝" SP		¾" SP		1" SP		2" SP		3" SP		4" SP		5" SP		6" SP	
		RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
11040	800	224	.74	245	.99	265	1.24	284	1.51	301	1.79	339	2.41	459	5.41	576	9.13	669	13.3	751	17.9	824	22.9
12420	900	241	.91	261	1.19	283	1.47	297	1.75	314	2.05	347	2.65	458	5.75	574	9.54	666	13.9	748	15.7	821	23.8
13800	1000	259	1.11	278	1.43	295	1.73	312	2.04	328	2.35	358	3.02	471	6.16	572	10.1	664	14.5	744	19.3	819	24.7
15180	1100	278	1.33	296	1.69	312	2.01	328	2.35	342	2.69	371	3.39	477	6.62	573	10.5	651	15.1	742	20.1	816	25.5
16560	1200	297	1.59	314	1.95	329	2.33	344	2.71	358	3.07	385	3.82	484	7.15	576	11.1	651	15.7	740	20.8	813	26.4
17940	1300	316	1.86	332	2.31	347	2.73	361	3.09	375	3.49	400	4.29	494	7.75	581	11.8	663	16.5	740	21.6	811	27.3
19320	1400	336	2.22	351	2.68	365	3.13	379	3.53	392	3.96	416	4.82	505	8.45	588	12.6	666	17.2	740	22.5	810	28.2
20700	1500	356	2.61	370	3.09	383	3.53	397	4.01	409	4.46	432	5.31	517	9.12	596	13.4	671	18.2	742	23.5	810	29.2
22080	1600	376	3.05	390	3.56	403	4.05	415	4.53	426	4.99	443	5.93	531	9.97	606	14.4	678	19.2	746	24.5	812	30.4
24840	1800	417	4.06	430	4.64	441	5.18	457	5.71	463	6.24	484	7.35	560	11.8	630	16.5	695	21.5	758	26.9	820	32.9
27600	2000	459	5.31	470	5.95	481	6.54	491	7.14	501	7.74	520	8.95	591	13.4	657	18.9	717	24.2	776	29.8	[illegible]	35.9
30360	2200	501	6.81	512	7.56	521	8.19	530	8.81	539	9.46	557	10.8	624	16.1	636	21.6	743	27.2	798	33.2	850	39.3
33120	2400	544	8.61	554	9.43	562	10.1	571	10.8	579	11.5	596	12.9	659	18.7	717	24.6	771	30.6	822	35.8	872	41.3
35880	2600	587	10.7	596	11.6	604	12.4	612	13.1	613	13.8	635	15.4	694	21.6	749	27.9	801	34.4	849	40.9	[illegible]	47.8
38640	2800	629	13.1	639	14.1	646	14.9	653	15.6	660	16.5	675	18.2	730	24.8	783	31.6	832	38.4	879	45.5	924	52.6
41400	3000	672	15.9	681	17.1	688	17.8	695	18.7	702	19.6	715	21.3	768	28.4	818	35.6	865	42.9	910	50.4	953	57.8
44160	3200	716	19.1	725	20.3	731	21.7	736	22.1	743	22.9	756	24.8	805	32.3	853	40.1	899	47.7	942	55.8	984	61.8
46920	3400	759	22.6	768	24.1	773	24.8	779	25.8	785	26.8	797	28.7	844	36.7	889	44.7	933	53.1	975	61.4	1015	69.9

BHP shown does not include belt drive loss.

HOW TO SELECT YOUR MAKE-UP AIR HEATE

FEATURES OF W. C. GRANT MA UNITS

Quick, Low Cost Installation

The MA unit is delivered as a complete package, including fan, motor operated shutter, completely wired controls and all piping. The heavy, reinforced housing is mounted on an integral, channel iron frame to assure rigidity, simplify installation and eliminate the cost of a field erected platform. Easy erection, simple connection of gas and electrical service and positioning of the temperature control sensing element complete the installation. Each unit is fired, adjusted, and completely tested before shipment. Field adjustment of the pressure regulator readies the MA unit for operation. No other adjustments are needed.

Economical — Complete Combustion

MA units are less expensive to purchase and operate than ordinary space heater make-up air systems. Combustion is so complete (up to 100% efficiency, compared to 80% maximum in other types) that gas concentrations are far below the maximums allowed by all federal, state and local regulations. Even if the unit were to be intentionally set out of adjustment, the products of combustion would be well below the maximum allowable concentrations as set forth by the American Conference of Governmental Industrial Hygienists.

Fail-Safe Controls

Flame protection, temperature, air flow and gas pressure controls not only govern the starting cycle but, also, stop the unit fail-safe in case of any malfunction, such as improper gas pressure, flame or power failure, excessive temperatures, or the failure of fan or motor.

Insurance Company Approved

Type MA units will meet all requirements of FM or FIA

Full Temperature Control

Burner provides full proportioning temperature control up to a 30:1 turn-down ratio.

Warm Weather Ventilation

A separate switch disconnects all gas circuitry, thus permitting the fan to be operated independently in warm weather.

Volume Requirements

For the proper operation of a ventilating system, the quantity of make-up air should exceed the exhausted air by 5% to 10%. This will help provide a positive pressure in the building. Consult local codes for requirements or restrictions.

Heating Requirements

After calculating the volume of make-up air required, select the temperature rise of that unit. The temperature rise is the difference between the lowest expected outside temperature and the desired delivery temperature.

System Static Pressure Requirements

This refers to equipment and accessories external to the basic blower-burner unit. The following static pressure allowances should be used for accessories to the W. C. Grant Company unit.

Fresh Air Intake Hood .... ⅛ in. W.C.
Filter Section .... ⅛ in. W.C.
Discharge Diffusers ...... ¼ in. W.C.

In addition to these, be certain to include allowances for any inlet and outlet duct work.

VOLUME

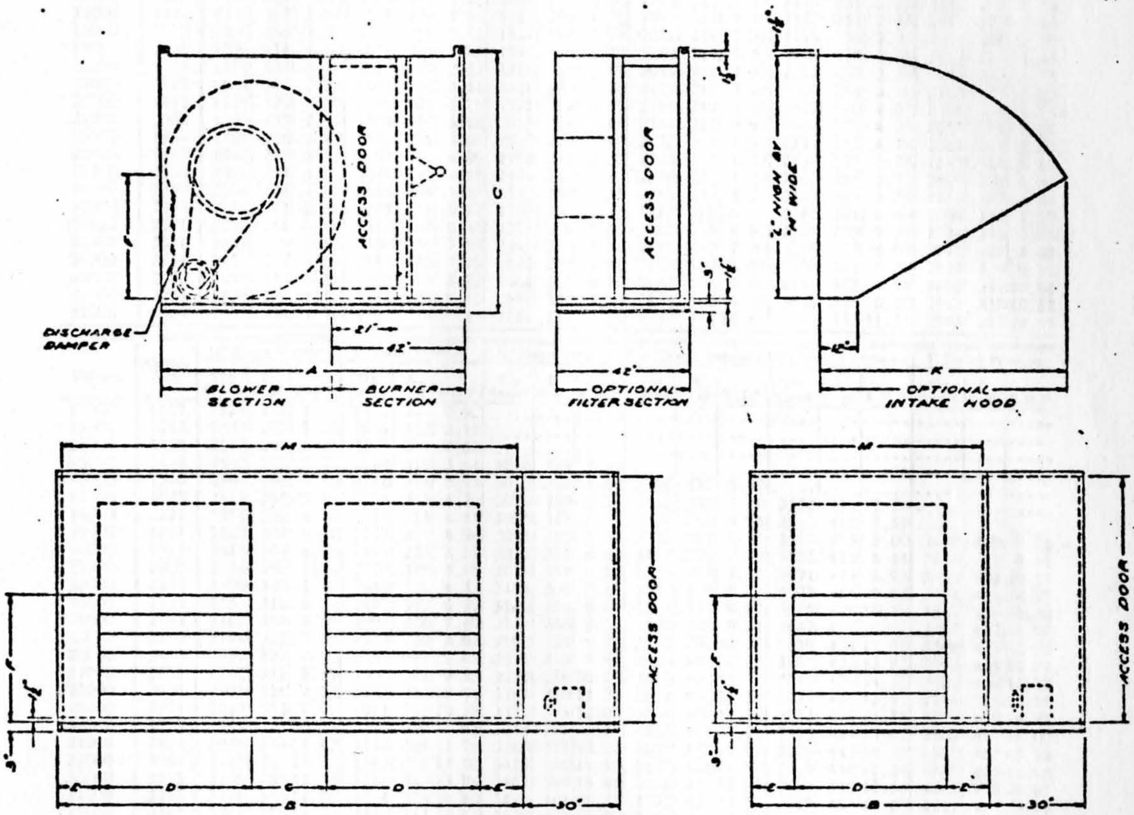
MODEL	CFM	DWDI BLOWER	OUTLET VELOCITY FPM	¼" SP RPM / BHP	½" SP RPM / BHP	¾" SP RPM / BHP	1" SP RPM / BHP
MA 10	10000	1-24	1820	505 / 4.05	555 / 4.94	592 / 5.79	622 / 6.60
MA 15	15000	1-30	1760	[illegible] / 5.34	443 / 6.26	476 / 7.16	498 / 8.2
MA 20	20000	1-30	2350	442 / 8.83	475 / 9.90	508 / 10.65	535 / 11.93
MA 25	25000	1-36	1970	354 / 9.37	384 / 10.57	412 / 12.00	438 / 13.52
MA 30	30000	1-36	2362	366 / 13.34	390 / 14.[illegible]	420 / 15.99	444 / 17.38
MA 35	35000	1-36	2760	362 / 19.02	407 / 19.48	432 / 21.54	456 / 22.81
MA 40	40000	2-30	2350	442 / 17.76	475 / 19.80	500 / 21.30	538 / 23.96
MA 45	45000	2-30	2645	434 / 21.84	466 / 24.36	518 / 26.50	547 / 28.16
MA 50	50000	2-33	2330	399 / 22.20	429 / 24.84	455 / 26.50	[illegible] / 29.16
MA 55	55000	2-33	2630	408 / 26.14	438 / 29.18	466 / 31.64	432 / 33.84
MA 60	60000	2-36	2362	365 / 25.73	396 / 28.76	420 / 31.93	444 / 34.9[illegible]
MA 65	65000	2-36	2562	373 / 30.70	400 / 34.32	42[illegible] / 37.20	450 / 39.56
MA 70	70000	2-36	276[illegible]	382 / 36.04	407 / [illegible]	432 / 43.08	455 / 45.60

TEMPERATURE RISE °F

MODEL	60° BTU/HR	70° BTU/HR	80° BTU/HR	90° BTU/HR	100° BTU/HR
MA 10	660.000	770.000	880.000	980.000	1.070.000
MA 15	990.000	1.155.000	1.320.000	1.470.000	1.605.000
MA 20	1.320.000	1.540.000	1.760.000	1.960.000	2.140.000
MA 25	1.650.000	1.925.000	2.200.000	2.450.000	2.675.000
MA 30	1.980.000	2.310.000	2.640.000	2.940.000	3.210.000
MA 35	2.310.000	2.695.000	3.080.000	3.430.000	3.745.000
MA 40	2.640.000	3.080.000	3.520.000	3.920.000	4.280.000
MA 45	2.970.000	3.465.000	3.960.000	4.410.000	4.815.000
MA 50	3.300.000	3.850.000	4.400.000	4.900.000	5.350.000
MA 55	3.630.000	4.230.000	4.840.000	5.390.000	5.885.000
MA 60	3.960.000	4.620.000	5.280.000	5.880.000	6.420.000
MA 65	4.290.000	5.000.000	5.720.000	6.370.000	6.955.000
MA 70	4.620.000	5.390.000	6.160.000	6.860.000	7.490.000

HORIZONTAL MAKE-UP AIR HEATERS

Horizontal models are best suited to buildings where on-roof, or under-roof (ceiling) installation is preferred. All weather maintenance accessibility, building construction and location may determine the selection of the horizontal unit.



MODEL	A	B	C	D	E	F	G	K	L	M	MAKE-UP AIR UNIT WEIGHT	FILTER SECTION WEIGHT	INTAKE HOOD WEIGHT
MA 10	78	51	56	31⅜	9½	25	—	55	50	48	920	310	60
MA 15	93	63	66	40 3/16	11½	30¾	—	67	60	60	1330	410	75
MA 20	93	63	66	40 3/16	11½	30¾	—	67	60	60	1550	440	90
MA 25	96	75	81	47 11/16	13½	38½	—	77	75	72	1600	440	90
MA 30	96	75	81	47 11/16	13½	38½	—	77	75	72	1900	495	110
MA 35	96	75	81	47 11/16	13½	38½	—	77	75	72	2600	615	150
MA 40	93	124	67	40 3/16	11½	30¾	20	67	60	121	3170	725	225
MA 45	93	124	67	40 3/16	11½	30¾	20	67	60	121	3210	725	225
MA 50	95	137	82	44 11/16	12½	34	22	77	75	134	3400	790	250
MA 55	95	137	82	44 11/16	12½	34	22	77	75	134	3425	790	250
MA 60	96	147	82	47 11/16	13½	38½	24	77	75	144	3525	790	250
MA 65	96	147	82	47 11/16	13½	38½	24	77	75	144	3550	790	250
MA 70	96	147	82	47 11/16	13½	38½	24	77	75	144	3600	790	250

Performance data — based on standard air at constant resistance.

NO. 33 TYPE FC - DOUBLE WIDTH - DOUBLE INLET FAN

With forwardly curved blade wheel — class 1 & 2

Volume	Outlet Vel.	¼" Static Pressure			⅜" Static Pressure			½" Static Pressure			⅝" Static Pressure			¾" Static Pressure			⅞" Static Pressure		
		Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.
8000	765	1029	119	.2[illegible]	1368	155	.49	1652	191	.69	1885	218	.97	....	....	....	....	....	....
9000	861	1063	123	.35	1381	160	.57	1661	192	.80	1894	219	1.08	....	....	....	....	....	....
10000	957	1099	127	.43	1402	162	.67	1670	193	.91	1903	220	1.20	2127	246	1.51	2319	263	1.89
11000	1052	1142	132	.54	1418	164	.79	1687	195	1.05	1920	222	1.34	2136	247	1.66	2328	269	2.03
12000	1147	1194	138	.65	1445	167	.92	1704	197	1.20	1935	224	1.50	2145	248	1.82	2337	270	2.18
13000	1243	1246	144	.80	1470	170	1.06	1730	200	1.37	1954	226	1.68	2154	249	2.01	2344	271	2.37
14000	1338	1307	151	.96	1506	174	1.23	1756	203	1.56	1972	228	1.88	2172	251	2.22	2353	272	2.58
15000	1434	1363	158	1.14	1549	179	1.42	1783	206	1.76	1998	231	2.10	2188	253	2.45	2371	274	2.83
16000	1530	1436	166	1.36	1592	184	1.62	1817	210	1.99	2024	234	2.35	2215	256	2.72	2386	276	3.10
17000	1625	1506	174	1.60	1635	189	1.85	1852	214	2.23	2050	237	2.63	2223	259	3.01	2413	279	3.42
18000	1721	1575	182	1.90	1687	195	2.11	1885	218	2.49	2085	241	2.92	2262	262	3.34	2440	282	3.78
19000	1817	1643	190	2.23	1748	202	2.41	1929	223	2.78	2115	245	3.24	2300	266	3.70	2465	285	4.20
20000	1912	1721	198	2.59	1799	208	2.76	1972	228	3.11	2154	249	3.55	2337	270	4.05	2491	288	4.55
21000	2008	1790	207	2.9[illegible]	1869	216	3.15	2015	233	3.48	2188	253	3.93	2362	273	4.2[illegible]	2517	291	4.94
22000	2104	1869	216	3.41	1938	224	3.66	2067	239	3.87	2233	258	4.31	2395	277	4.79	2552	295	5.38
23000	2199	1947	225	3.85	2007	232	4.09	2118	245	4.32	2266	262	4.73	2430	281	5.23	2586	299	5.82
24000	2295	2024	234	4.37	2075	240	4.58	2180	252	4.80	2310	267	5.2[illegible]	2465	285	5.72	2620	303	6.31
25000	2390	2104	243	4.91	2154	249	5.13	2223	259	5.35	2362	273	5.7[illegible]	2499	289	6.25	2656	307	6.84
26000	2485	2180	252	5.51	2233	258	5.72	2300	266	5.93	2413	279	6.2[illegible]	2542	294	6.80	2690	311	7.39
27000	2581	2257	261	6.15	2310	267	6.36	2371	274	6.58	2465	285	6.92	2586	299	7.39	2734	316	7.98
28000	2678	2337	270	6.82	2386	276	7.03	2447	283	7.23	2525	292	7.66	2638	305	8.02	2777	321	8.60
29000	2774	2413	279	7.57	2474	286	7.79	2525	292	7.98	2595	300	8.31	2690	311	8.78	2820	326	9.31
30000	2870	2491	288	8.33	2560	296	8.53	2604	301	8.72	2663	308	9.06	2751	318	9.51	2862	331	10.03
31000	2964	2579	298	9.12	2638	305	9.32	2681	310	9.52	2734	316	9.85	2811	325	10.32	2914	337	10.82
32000	3060	2663	308	9.96	2717	314	10.15	2751	318	10.33	2802	324	10.6[illegible]	2880	333	11.14	2965	343	11.65

Volume	Outlet Vel.	⅞" Static Pressure			1" Static Pressure			1¼" Static Pressure			1½" Static Pressure			1¾" Static Pressure			2" Static Pressure		
		Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.
12000	1147	2525	292	2.55	2674	309	2.92	....	....	....	....	....	....	....	....	....	....	....	....
13000	1243	2533	293	2.75	2681	310	3.14	....	....	....	....	....	....	....	....	....	....	....	....
14000	1338	2542	294	2.98	2690	311	3.38	....	....	....	....	....	....	....	....	....	....	....	....
15000	1434	2552	295	3.22	2699	312	3.64	3010	348	4.53	....	....	....	....	....	....	....	....	....
16000	1530	2560	296	3.50	2716	314	3.94	3018	349	4.81	3304	382	5.92	....	....	....	....	....	....
17000	1625	2579	298	3.81	2734	316	4.24	3026	350	5.15	3313	383	6.20	3553	411	7.26	....	....	....
18000	1721	2595	300	4.14	2751	318	4.56	3035	351	5.54	3321	384	6.56	3562	412	7.59	....	....	....
19000	1817	2620	303	4.48	2777	321	4.90	3053	353	5.97	3330	385	6.95	3570	413	8.00	3807	440	9.25
20000	1912	2646	306	4.89	2794	323	5.39	3071	355	6.42	3339	386	7.40	3582	414	8.44	3816	441	9.67
21000	2008	2674	309	5.34	2820	326	5.86	3088	357	6.91	3348	387	7.91	3591	415	8.95	3826	442	10.17
22000	2104	2699	312	5.81	2845	329	6.33	3115	360	7.42	3365	389	8.43	3607	417	9.52	3834	443	10.70
23000	2199	2734	316	6.32	2872	332	6.92	3140	363	8.00	3383	391	9.02	3625	419	10.1[illegible]	3849	445	11.30
24000	2295	2760	319	6.86	2907	336	7.49	3166	366	8.58	3407	394	9.64	3642	421	10.6[illegible]	3867	447	11.98
25000	2390	2794	323	7.44	2932	339	8.07	3192	369	9.20	3435	397	10.35	3670	424	11.56	3886	449	12.68
26000	2485	2827	327	7.98	2965	343	8.69	3225	373	9.87	3460	400	11.01	3686	426	12.32	3903	451	13.44
27000	2581	2862	331	8.59	3000	347	9.35	3252	376	10.55	3485	403	11.77	3713	429	13.10	3929	454	14.27
28000	2678	2907	336	9.23	3033	351	10.04	3287	380	11.23	3512	406	12.56	3738	432	13.9[illegible]	3943	456	15.16
29000	2774	2940	340	9.92	3071	355	10.78	3321	384	12.00	3546	410	13.38	3762	435	14.75	3972	459	16.05
30000	2870	2984	345	10.63	3105	359	11.51	3348	387	12.78	3582	414	14.21	3800	439	15.62	3998	462	17.00
31000	2964	3026	350	11.42	3147	364	12.75	3383	391	13.62	3607	417	15.07	3826	442	16.57	4022	465	17.98
32000	3060	3071	355	12.25	3182	368	13.10	3418	395	14.50	3642	421	15.95	3859	446	17.54	4045	468	18.98
33000	3155	3115	360	13.14	3225	373	13.98	3452	399	15.41	3678	425	16.87	3886	449	18.55	4083	472	20.01
34000	3252	3166	366	14.03	3268	378	14.89	3499	404	16.34	3702	428	17.85	3917	453	19.66	4111	475	21.10
35000	3347	3217	372	14.96	3321	384	15.82	3529	408	17.28	3738	432	18.89	3943	456	20.67	4144	479	22.22
36000	3442	3279	379	15.9[illegible]	3365	389	16.8[illegible]	3570	413	18.31	3769	436	19.97	3978	460	21.78	4179	483	23.41

Volume	Outlet Vel.	2¼" Static Pressure			2½" Static Pressure			3" Static Pressure			3½" Static Pressure			4" Static Pressure			5" Static Pressure		
		Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.	Tip Speed	R. P. M.	B.H.P.
21000	2008	4045	468	11.33	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
22000	2104	4054	469	11.86	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
23000	2199	4063	470	12.45	4281	495	13.86	....	....	....	....	....	....	....	....	....	....	....	....
24000	2295	4071	471	13.16	4290	496	14.54	4668	540	17.44	....	....	....	....	....	....	....	....	....
25000	2390	4080	472	13.97	4299	497	15.2[illegible]	4677	541	18.14	....	....	....	....	....	....	....	....	....
26000	2485	4098	474	14.73	4308	498	15.04	4685	542	18.88	....	....	....	....	....	....	....	....	....
27000	2581	4115	476	15.58	4325	500	16.86	4694	543	19.7[illegible]	5052	584	22.79	....	....	....	....	....	....
28000	2678	4142	479	16.44	4343	502	17.73	4711	545	20.63	5061	585	23.56	5400	624	26.93	....	....	....
29000	2774	4159	481	17.4[illegible]	4360	504	18.66	4720	546	21.60	5070	586	24.64	5409	625	27.94	6036	698	35.39
30000	2870	4187	484	18.39	4377	506	19.67	4736	548	22.61	5078	587	25.72	5418	626	29.03	6045	699	36.34
31000	2964	4215	487	19.38	4402	509	20.75	4762	551	23.70	5095	589	26.82	5426	627	30.18	6054	700	37.40
32000	3060	4237	490	20.45	4428	512	21.87	4788	554	24.83	5112	591	28.03	5435	628	31.39	6062	701	38.59
33000	3155	4263	493	21.66	4454	515	23.05	4812	556	26.10	5129	593	29.35	5444	629	32.60	6071	702	39.83
34000	3252	4290	496	22.77	4482	518	24.25	4836	559	27.39	5147	595	30.65	5461	631	33.87	6088	704	41.20
35000	3347	4325	500	23.9[illegible]	4509	521	25.52	4863	562	28.74	5170	598	32.02	5487	634	35.21	6105	706	42.63
36000	3442	4360	504	25.20	4534	524	26.77	4887	565	30.07	5188	600	33.43	5504	636	36.70	6122	708	44.12
37000	3538	4387	507	26.43	4561	527	28.04	4911	568	31.5[illegible]	5214	603	34.91	5521	638	38.26	6139	710	45.65
38000	3634	4416	511	27.72	4596	531	29.41	4937	571	32.90	5240	606	36.49	5547	641	39.91	6156	712	47.30
39000	3730	4445	514	29.05	4630	535	30.81	4963	574	34.41	5266	609	37.99	5563	643	41.62	6173	714	49.02
40000	3824	4482	518	30.42	4643	538	32.26	4998	578	35.94	5292	612	39.68	5599	647	43.43	6190	716	50.81
41000	3920	4516	522	31.56	4683	542	33.77	5026	581	37.61	5327	616	41.43	5615	650	45.30	6207	718	52.68
42000	4016	4550	526	33.35	4718	546	35.30	5052	584	39.32	5354	619	43.36	5641	653	47.21	6233	721	54.58

BARRY BLOWER COMPANY — MINNEAPOLIS, MINN.

TABLE 7

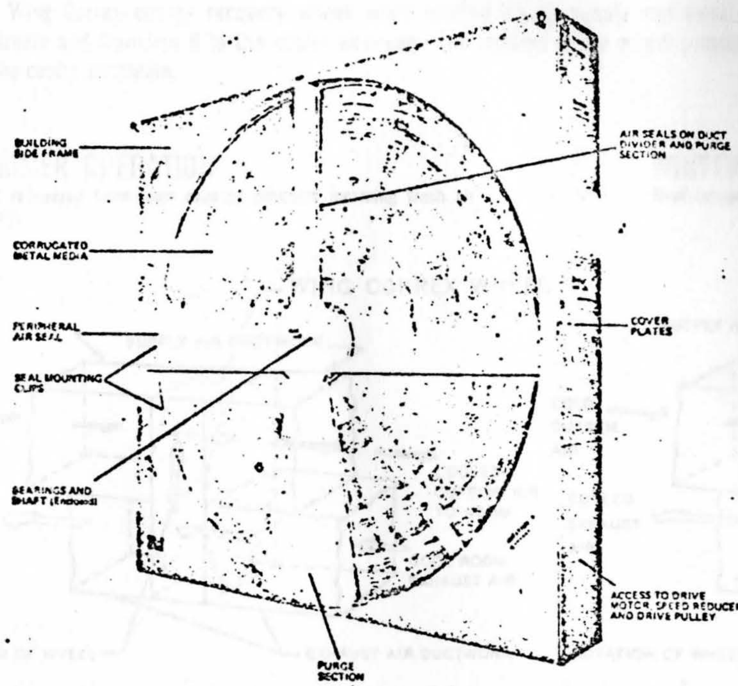
DESCRIPTION OF TANK SOLUTIONS IN ANODIZING LINE

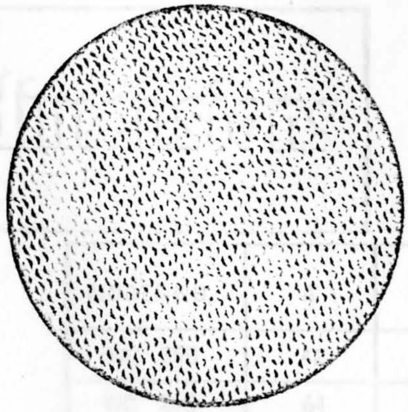
Tank Number	Solution	Temperature	Heat Loss (BTU/hr)	Tank Size (gallons)
1	Alkaline Clean	160°F	227,852	3656
2	Rinse	ambient	---	3656
3	Rinse	ambient	---	3656
4	Acid Clean	170°F	268,324	3656
5	Etch	140°F	155,536	6094
6	Rinse	ambient	---	3656
7	Rinse	ambient	---	3656
8	Bright Dip	200°F	453,042	3656
9	Rinse	ambient	---	3656
10	Rinse	ambient	---	3656
11	Rinse	ambient	---	3656
12	Desmut	ambient	---	3656
13	Rinse	ambient	---	3656
14	Anodize	ambient	---	3656
15	Anodize	ambient	---	3656
16	Rinse	ambient	---	3656
17	Rinse	ambient	---	3656
18	Color	150°F	139,875	3656
19	Color	145°F	131,540	3656
20	Color	150°F	139,875	3656
21	Rinse	ambient	---	3656
22	Color	140°F	114,730	3656
23	Color	150°F	139,875	3656

TABLE 7 (CONT.)

Tank Number	Solution	Temperature	Heat Loss (BTU/hr)	Tank Size (gallons)
24	Color	135°F	111,370	3656
25	Color	150°F	139,875	3656
26	Nickel Acetate	160°F	227,852	3656
27	Rinse	ambient	---	3656
28	Rinse	ambient	---	3656
29	Hot Water Seal	200°F	453,042	3656
30	Hot Water Seal	200°F	453,042	3656

CORREX — ROTARY AIR TO AIR SENSIBLE HEAT EXCHANGER




CLOSEUP OF WHEEL SURFACE

Can recover up to 85% of wasted sensible energy in exhaust airstream and uses it to preheat or pre-cool the incoming air stream in makeup air heating and ventilating systems in all buildings requiring introduction of outside air to replenish that which has been exhausted.

This reduces operating costs of heating makeup air in the wintertime and reduces operating costs of cooling makeup air in the summer.

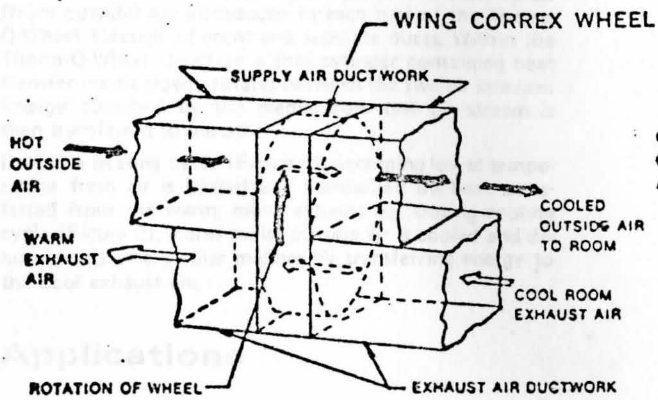
It reduces the size and costs of boiler and refrigeration equipment and will permit expansion of existing facilities without enlarging existing boiler or refrigeration plants.

operation

The Wing Correx energy recovery wheel, when rotating across supply and exhaust ductwork, absorbs heat from the warmer airstream and transfers it to the cooler airstream. The rotation of the wheel provides a "flow" of heat energy from the warmer to the cooler airstream.

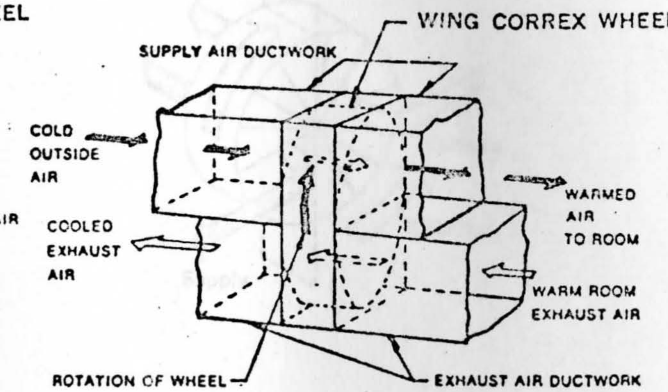
SUMMER OPERATION

Cold recovered from room exhaust precools incoming fresh air supply.

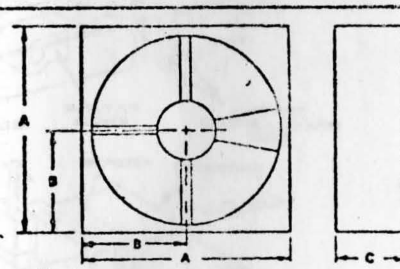


WINTER OPERATION

Heat recovered from room air preheats incoming fresh air supply.



dimensions



MODEL NO.	A	B	C	FACE AREA	NET WEIGHT LBS.
WC 350	42	21	12	3.30	600
WC 875	64	32	13⅝	8.13	1150
WC 1400	83	41½	13⅝	13.15	2150
WC 2150	101	51	16⅝	21.10	2450
WC 3160	120	60	19⅝	31.00	3100
WC 4770	144	72	19⅝	46.05	4350

Principle of Operation

Carnes Therm-O-Wheel represents a highly efficient method of transferring valuable heat energy contained in exhaust air streams. Energy is retained by transferring the heating or cooling effect from the exhaust air stream to the incoming outside air stream.

Unique in its design and function, the Carnes Therm-O-Wheel has the ability to transfer both moisture and temperature (latent and sensible heat) to provide uniform supply air conditions. Highly efficient and compact, the Carnes Therm-O-Wheel provides for year-round environmental control while reducing make-up air loading.

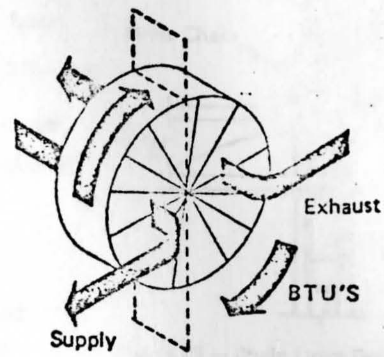
Ventilation air is generally required by code, comfort, or process. Introduction of fresh air to these spaces helps maintain design conditions by dilution and purging.

Systems exhausting large quantities of preconditioned air (or gas) represent outside air thermal polution and wasted energy. Conditioning fresh air for make-up places a substantial load on the air conditioning or heating system.

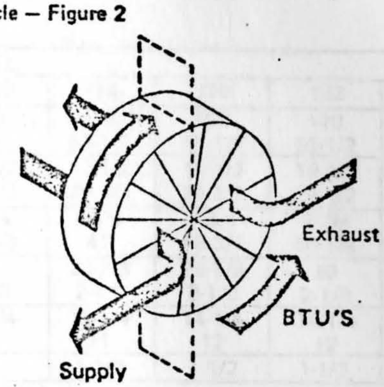
Exhaust air (from the building or space) and make-up air (from outside) are introduced to each half of the Therm-O-Wheel through adjacent and separate ducts. Within the Therm-O-Wheel structure a thin cylinder containing heat transfer media slowly rotates between the two air streams. Energy absorbed by the media from one air stream is then transferred to the other.

During a heating cycle (Figure 1), incoming lower temperature fresh air is heated and humidified by heat transferred from the warm, moist exhaust air. During cooling cycle (Figure 2), warm moist outside air is cooled and dehumidified in a similar manner by transferring energy to the cool exhaust air.

Heating Cycle — Figure 1



Cooling Cycle — Figure 2

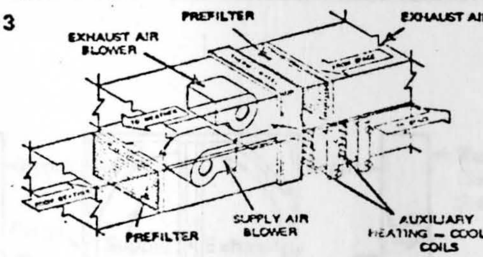


Applications

Central Air Conditioning Systems (Figure 3)

The Therm-O-Wheel when used in a comfort application, allows for tempering of the outside air on a year-round basis when it is coupled with a heating and cooling system. Reduction of fuel and power cost is realized when the Therm-O-Wheel heats and humidifies outside winter air and cools and dehumidifies outside summer air.

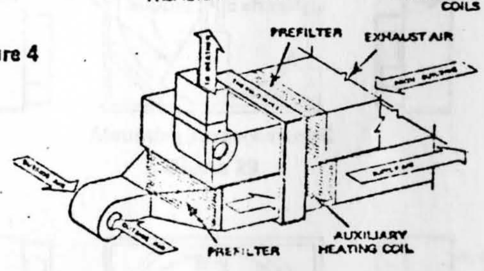
Figure 3



Industrial Make-Up Air Systems (Figure 4)

The cost of heating a factory or any other large industrial complex can be greatly reduced with an installation of a Carnes Therm-O-Wheel. The Therm-O-Wheel allows for additional exhaust and supply air volumes which relieve negative pressures and eliminates industrial pollutants from within the conditioned space.

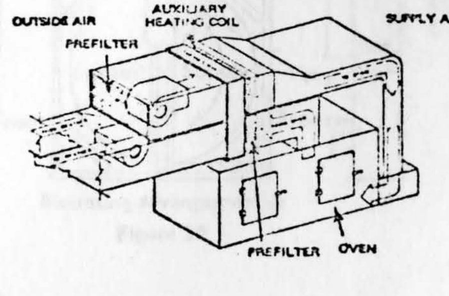
Figure 4



Industrial Processes (Figure 5)

With the installation of a Therm-O-Wheel, fuel costs will be significantly reduced for drying or baking oven applications. The unit uses the exhaust air to preheat the supply air going back to the process or air for other processes. Heating air from these processes can also be used for pre-heating outside air for comfort applications within a conditioned space.

Figure 5



Dimensional Data

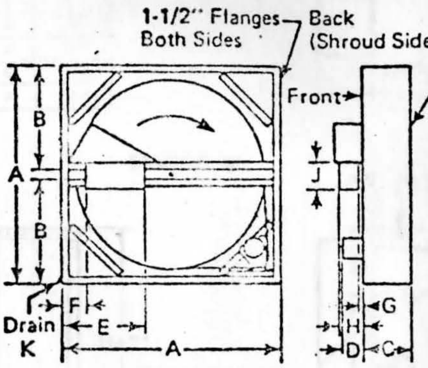


Figure 24

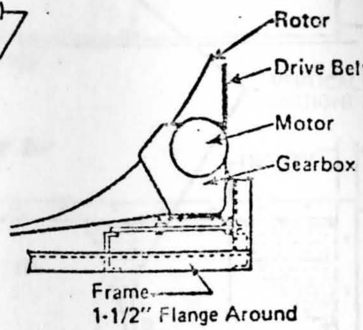


Figure 25 – Belt Drive Detail "PS" and "MS"

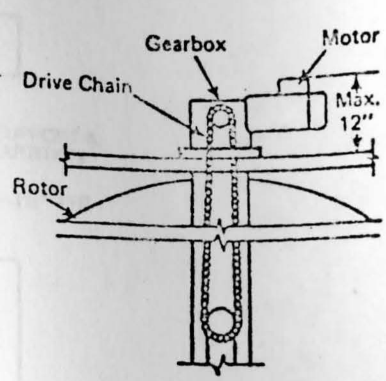


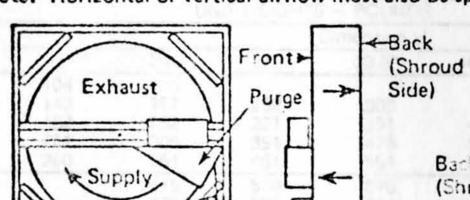
Figure 26 – Chain Drive Detail "HN"

TABLE 2

Dimensions (Inches)	MODEL								
	TO48	TO60	TO74	TO88	TO99	110	120	132	144
A	54	66	81	95	106	118	128	140	152
B	24-1/2	30	37-1/2	44-1/2	49	54-1/2	59-1/2	65-1/2	71-1/2
C (PS)	15-1/2	15-1/2	15-1/2	15-1/2	17-1/2	17-1/2	19-1/2	19-1/2	21-1/2
C (MS and HN)	201/2	20-1/2	20-1/2	20-1/2	22-1/2	22-1/2	24-1/2	24-1/2	26-1/2
D (Max.)	9-3/4	9-3/4	9-3/4	9-3/4	8-3/4	8-3/4	7-3/4	7-3/4	6-3/4
E	18-1/4	22-1/4	28-1/4	32-3/8	38-1/8	41	46-3/4	50-1/8	53-3/8
F	10-1/8	14-1/8	16-1/8	20-1/4	22	24-7/8	26-5/8	30	33-1/4
G	2-1/8	2-1/8	2-1/8	2-1/8	2-1/8	2-1/8	2-1/8	2-1/8	1-1/8
H	8-1/4	8-1/4	10-1/4	10-1/4	12-1/4	12-1/4	14-1/4	14-1/4	13-1/4
J	8	8	9	9	11	11	12	12	12
K (FPT Size)	3/4	3/4	1	1	1	1-1/2	1-1/2	1-1/2	1-1/2
Net Weight (Lbs.)									
[illegible]	790	935	1155	1390	1730	1935	2370	2850	3335
MS	835	1000	1250	1525	1900	2200	2625	3100	3700
HN	1000	1265	1625	2000	2460	2930	3450	4050	4850

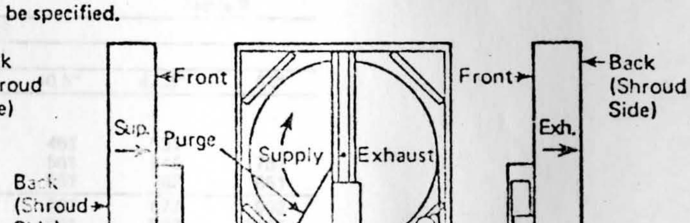
Mounting Arrangement

Note: Horizontal or vertical airflow must also be specified.



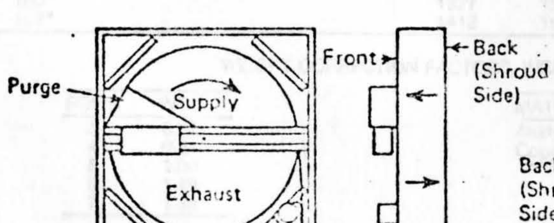
Mounting Arrangement A

Figure 27



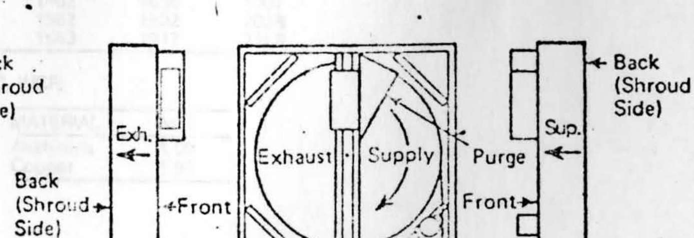
Mounting Arrangement C

Figure 29



Mounting Arrangement B

Figure 28



Mounting Arrangement D

Figure 30

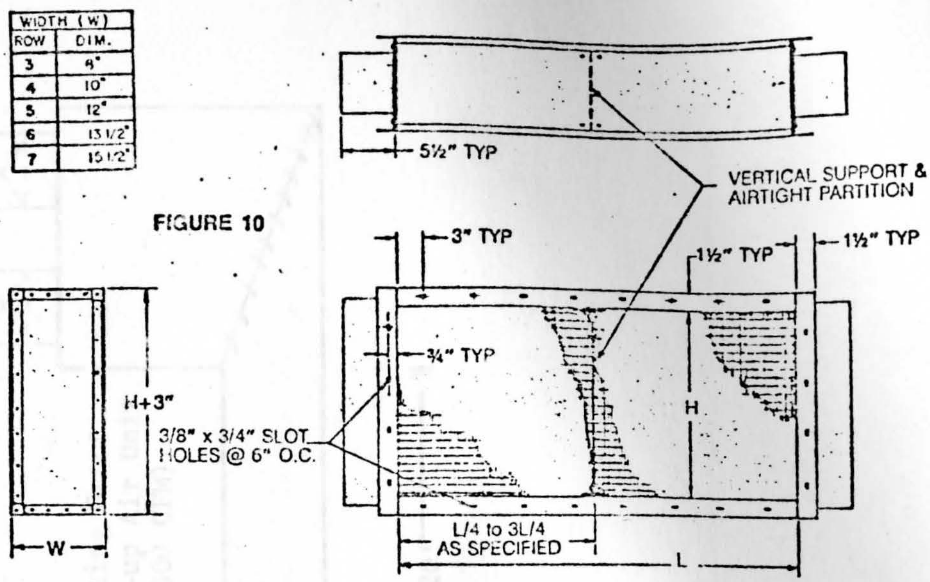


FIGURE 10

Q-PIPE® UNIT DIMENSIONS — Table 1

Dimension L	UNIT DIMENSIONS AND TOTAL FACE AREA ~ SQ. FT. Dimension H 13.5"	20.3"	27"	33.8"	40.5"	47.3"	54"
24"	2.3	3.4					
36"	3.4	5.1	6.8	8.5			
48"	4.5	6.8	9.0	11.3	13.5	15.8	
60"	5.6	8.5	11.3	14.1	16.9	19.7	22.5
72"	6.8	10.2	13.5	16.9	20.3	23.7	27.0
84"		11.8	15.8	19.7	23.6	27.6	31.5
96"		13.5	18.0	22.5	27.0	31.5	36.0
108"		15.2	20.3	25.4	30.4	35.5	40.5
120"			22.5	28.2	33.8	39.4	45.0
132"			24.8	31.0	37.1	43.4	49.5
144"			27.0	33.8	40.5	47.3	54.0
156"			29.3	36.6	43.9	51.2	58.5
168"				39.4	47.3	55.2	63.0
180"				42.3	50.6	59.1	67.5
192"				45.1	54.0	63.1	72.0

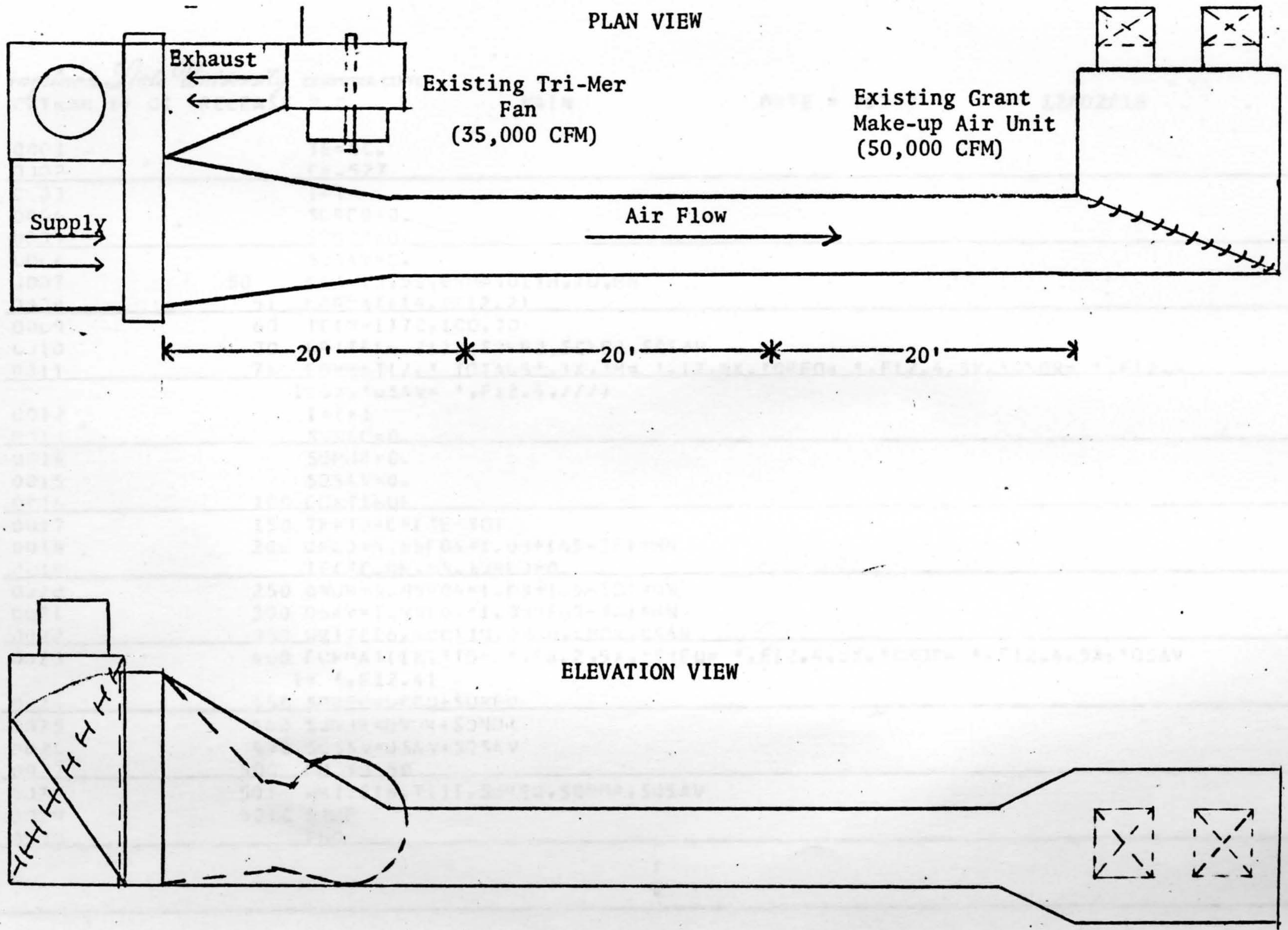
Q-PIPE® UNIT WEIGHTS (5 row, 14 fpi) — Table 2

Dimension L	UNIT WEIGHTS ~ POUNDS Dimension H 13.5"	20.3"	27"	33.8"	40.5"	47.3"	54"
24"	104	143					
36"	143	197	252	306			
48"	182	252	321	391	461	531	
60"	221	306	391	476	561	646	731
72"	260	361	461	561	661	762	861
84"		415	530	646	761	877	992
96"		470	600	731	861	993	1123
108"		525	669	816	961	1103	1253
120"			739	902	1062	1224	1384
132"			809	987	1162	1339	1515
144"			878	1072	1262	1455	1645
156"			948	1157	1362	1571	1776
168"				1242	1462	1686	1907
180"				1327	1562	1802	2038
192"				1412	1663	1917	2168

WEIGHT CORRECTION FACTORS, WCF:

ROWS	WCF
3	0.62
4	0.81
5	1.00
6	1.19
7	1.38

MATERIAL	WCF
Aluminum	1.00
Copper	1.96


PLAN VIEW
Exhaust
Existing Tri-Mer
Fan
(35,000 CFM)
Existing Grant
Make-up Air Unit
(50,000 CFM)
Supply
Air Flow
20'
20'
20'
ELEVATION VIEW

```
                TE=80.
                E=.527
                I=1
                SQREQ=0.
                SQNOR=0.
                SQSAV=0.
          50    READ(5,51,END=501)M,TO,HN
           51   FORMAT(I4,2F12.2)
           60   IF(M-1)70,100,70
           70   WRITE(6,71)I,SQREQ,SQNOR,SQSAV
           71   FORMAT(/,' TOTALS',3X,'M= ',I2,5X,'QREQ= ',E12.4,5X,'QNOR= ',E12.4
               1,5X,'QSAV= ',E12.4,///)
                I=I+1
                SQREQ=0.
                SQNOR=0.
                SQSAV=0.
          100   CONTINUE
          150   TF=TO+E*(TE-TO)
          200   QREQ=4.85E04*1.08*(65-TF)*HN
                IF(TF.GE.65.)QREQ=0.
          250   QNOR=4.85E04*1.08*(65-TO)*HN
          300   QSAV=1.93E04*1.08*(65-TO)*HN
          350   WRITE(6,400)TO,QREQ,QNOR,QSAV
          400   FORMAT(1X,'TO= ',F6.2,5X,'QREQ= ',E12.4,5X,'QNOR= ',E12.4,5X,'QSAV
               1= ',E12.4)
          450   SQREQ=QREQ+SQREQ
          460   SQNOR=QNOR+SQNOR
          470   SQSAV=QSAV+SQSAV
          500   GO TO 50
          501   WRITE(6,71)I,SQREQ,SQNOR,SQSAV
         5010   STOP
                END
```