Evaluation of an All-Glass, Evacuated, Tubular,
Nonfocusing, Nontracking Solar Collector Array

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ABSTRACT

A spaced, evacuated tubular radiation receiver employing a highly selective absorber surface and light enhancement features has been subject to extensive test and analysis. Air was used as the heat transfer fluid. The design, installation, test and analysis of the air cooled collector was conducted with the sponsorship of the Research and Development Branch, Solar Division, of the Energy Research and Development Administration under Contract No. EY-76-C-02-2919 000. The thermal performance of the air cooled collector was found to be better than predicted when rated on an all day operating basis. The high performance characteristic is enhanced by the features of ease of installation and freedom from maintenance and operational problems. High quality air or liquid may be provided (the latter by heat exchange) for a variety of load applications.

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I. Introduction

A. General

The economics and cost effectiveness of solar thermal collectors may be enhanced significantly if they are able to drive cooling cycles effectively as well as provide for space and domestic hot water heating applications. Higher temperatures introduce the potential for wider usage of solar thermal energy in providing heat for industrial processes, desalination, agriculture and food processing and similar frequently discussed energy dependent enterprises. The use of air as the energy transport fluid is becoming a preferred solution to many solar energy system practitioners.

Solar thermal energy collectors, using the spaced evacuated tubular absorber configuration, have demonstrated the ability to provide high grade solar energy. An understanding of the tubular collector configuration and the differences from and similarities to the traditional flat surface collector family is important to the user. The use of vacuum to eliminate the thermal losses due to conduction and convection, in conjunction with a highly selective absorber surface results in a low loss coefficient collector. The exploitation of the symmetry of the absorber surface through the spacing of the elements in conjunction with a simple diffuse light reflecting background enhances the effectiveness of each of the cost sensitive units. The light enhancement factor combined with the low loss coefficient characteristic improves the effectiveness of the use of air as the heat transfer or collector cooling fluid.

A one hundred forty-four (144) evacuated, spaced tubular air driven collector array was designed, installed and tested to evaluate and

demonstrate the effectiveness of an air cooled collector. Two additional arrays of equal size were also installed and tested. These arrays used a liquid as the heat transfer fluid. In one case the fluid was contained within the glass envelope of the absorber tube and in the other within formed metal tubing having an elongated U shape which was inserted within the absorber tube glass envelope. The test and evaluation effort was partially funded by the Solar Division of the Energy Research and Development Administration. The thermal performance and operating characteristics of the air collector array was explored over a range of fluid mass flow rates. The testing was conducted using data collection and evaluation on an all day basis. This allowed a considerable degree of flexibility in the design of the test, in the accuracy of the control required for the fluid mass flow rate, collector inlet temperature and the fluctuation of solar insolation throughout the daily test period. The latter is a very important consideration, at least in Toledo, Ohio, where clear day conditions are indeed a rarity.

All three collectors demonstrated a high level of thermal performance throughout an operating temperature regime from ambient to collector fluid discharge temperatures in the range of 300°F in the case of the air system and 250°F for the liquid systems, the latter temperature limit imposed by fluid pressure considerations. The performance was essentially unaffected by ambient conditions where temperatures of 15°F below zero and chill factors of 50°F below zero were experienced. Heavy and extended periods of snow fall in addition to high winds and extended cold were common during the 1976-1977 winter season in Toledo; the collectors were operated with no significant adverse affects due to such climatic conditions.

B. Summary Comments

1. Many companies/suppliers are now pursuing the selectively coated evacuated tubular collector as the receiver for solar thermal energy systems. The test and analysis program conducted through the present period of performance of the subject contract has demonstrated the need to develop a more thorough understanding of the various parameters which influence the performance of such receivers. Testing procedures as now called for by ASHRAE 93-77 would appear to require modification to apply to the spaced evacuated tubular configuration. The thermal mass and the fluid transit time must be considered for liquid systems. The long time constant which obtains when air is used as the heat transfer fluid has a significant influence on the test and analysis procedures which may be used. The present period of performance has allowed the acquisition of sufficient experimental data to verify the fundamental high performance characteristics of the evacuated tubular receiver element.

2. Several inconsistencies in the data require further exploration to resolve them. Analytical procedures had predicted an order of performance for the spaced receivers which ranked the liquid collector first followed by the air system and the stagnant air system in that order. On the other hand, the experimental data would rank the air systems as the highest performer on an all day basis near the ambient temperature intercept with the liquid and stagnant air liquid decreasing in rank in respective order. The loss terms, as indicated by the slope of the stagnant air liquid system, is the least and that of the air system the greatest.

One possible explanation for the inconsistency in the intercept ranking is the lack of experimental performance data near the intercept for both liquid systems. As indicated in the discussion, the lack of such data resulted from an inadequate heat dump capacity which prevented the liquid inlet temperatures from being clamped near to the ambient temperature all through the day. Another possible explanation is the heat capacity of the liquid collectors coupled with the transit time required for an element of fluid to pass through the collector. The method used for the analysis of thermal performance requires further study. The experimentally determined slope values, while significantly different numerically, are not greatly at variance in absolute magnitude. The loss coefficient is very low in all cases. Again, additional thermal performance data near the intercept could modify the slope values to quite an extent.

3. The installation and operation of the air collector proved to be trouble free. Attention to detail resulted in an air manifold which was free from air leakage. The collector tube elements were removed frequently for demonstration and/or inspection during any part of the operating day without incident. Heavy snow fall and extreme cold did not impact on the collector operation or to a measureable degree on collector performance. The collector was allowed to stagnate several times with air flow start up induced at peak temperature conditions. No sign of stress to the collector could be noted. The collector was often allowed to stagnate over week ends and periodically for extended periods. There was no evidence of any detrimental affect.

C. Conclusions

1. The use of air as the heat transfer fluid in a solar thermal collector causes no significant degradation in thermal performance when such use is in conjunction with a low loss coefficient receiver incorporating radiation enhancement features.

2. The use of air simplifies the packaging, installation, operation and maintenance of a solar collector. The problems which must be accommodated when liquids are used such as freezing, boiling, leakage and attendant/system shut down and/secondary damage, thermal shock, erosion/corrosion, potable water contamination, pressurization and attendant code considerations including safety and similar factors are completely avoided by the use of air.

3. The very low loss coefficient and radiation enhancement features allows the use of relatively low air mass flow rates resulting in very low pumping power requirements.

4. The same features allows the addition of an air-liquid heat exchange unit (where the load circuit elements dictate liquids) without a significant degradation in performance even when, for cost reasons, a low effectiveness (a 50%) heat exchanger is employed.

5. The necessity for heat dump or operational shut down of the collector to avoid over temperature/pressure conditions is minimized. The collector and manifold elements can operate continuously up to 350°F exit fluid temperatures at relatively high levels of efficiency. The reduced need for energy dump should enhance the thermal efficiency of the system. The higher temperature capability could impact significantly on storage element size and cost.

D. Recommendations

1. The present period of performance has added greatly to the understanding of the performance characteristics of the evacuated tubular collector and radiation enhancement techniques. A test bed having great versatility in the variation of operating parameters has been created. Analysis techniques have been developed. It is recommended and urged that the contract period of performance be extended for twelve (12) months in order to fill the gaps in the experimental data and to resolve the inconsistencies which now exist in the quantitative characterization of the evacuated tubular collector.

2. It is recommended that consideration be given to the development of solar energy systems utilizing the high performance air cooled evacuated tubular receiver. Specific unsolicited proposals for the systems integration of the air collector will be submitted under separate cover.

3. The need for the rating of solar collectors to a national standard culminated in the issuance of the ASHRAE 93-77 procedure. The test standard did not include suitable consideration to the wide variety of collectors currently being offered or under development for potential public sale. Specifically, the impact of all day performance under varying fractions of diffuse radiation was not included. The extended all day performance testing and analysis during the present period of performance of the subject contract surfaced many of the factors which must be considered in the rating of collectors. The cost of such extended testing and analysis

could be prohibitive if required for all potential collector manufacturers. However, the availability of all day test data in depth is necessary to qualify a shorter term test procedure. It is recommended that the proposed extension to the period of performance include provisions for the subcontract to a third party for testing and analysis on an extended basis. The preferred test site would be one such as the Desert Sunshine Exposure Test, Inc., or equivalent where all of the necessary instrumentation, data acquisition and analysis systems are in existence. Other important criteria are the climatic features where a predominance of clear day radiation conditions could be anticipated. The first emphasis in such a subcontract would be the correlation of long term test data with that obtained and to be obtained under the proposed extension in the period of performance. The second emphasis would be towards the development of a short term test procedure for the rating of the evacuated tubular type (and by inference all types) of solar energy collectors which would provide a suitable degree of correlation to the results of the long term test and analysis program. A specific unsolicited proposal for the recommended and contract will be submitted under separate cover.

II. Descriptions of the air, liquid and hybrid collector arrays.

A. Air collector array.

The air collector array consists of 144 collector tube elements connected into two manifolds of 72 elements each. Each module is approximately 12 feet long by 8 feet high. The collector tube elements are spaced on 4 inch centers to optimize the energy collection per total installed area. The modules are mounted one above the other as indicated in Figures 1 and 2. The array is south facing at a 41° latitude and at a tilt angle of 45° . The tubes are supported at their closed ends in simple sheet metal brackets using a plastic cup and light weight spring to maintain position. The open ends of the tubular elements are attached to the manifold through silicone O ring seals. No special tubular restraint system is required since the system operates at 0.5 to 1.0 inches of water gage pressure which imposes no significant separating or outward force on the tube elements and support structure.

Each manifold requires a roof penetration to provide connection to the distribution ducting mounted underneath the roofline. In the test array, the manifolds are center fed with the inlet and exhaust air ducting extending to the manifold ends and accommodating 36 tubes on either side of the feed plane. The major portion of the air manifold is hidden below the diffuse light reflecting backing screen. Extensions above the main manifold flow path accommodate the air flow to and from the collector tube elements. This configuration was chosen for aesthetic purposes only since the backing screen is not weather tight; the roof itself could have been painted white to provide the diffuse light reflecting surface necessary for the optimum thermal performance of the array.

The ducting under the roofline is contained within the roof support stringers. A central duct connects to the spur ducts to provide the flow path to and from an air blower. An air mass flow sensor and electric heater element are mounted integrally within the central ducting.

The test loop which was installed is shown schematically in Figure 3.

Ducting provides for full ingestion from and overboard dumping and to ambient air. A bypass and suitable dampers allow for up to full recirculation of the collector discharge air. An electric air preheater provides for approximate control of the collector inlet air temperature. A variable speed drive is used to control the air fan to provide air flow rates up to approximately 10 SCFM per tube pair.

The test system was extensively instrumented with temperature sensors. Four element thermopiles are located in the inlet and exit ducts of each of the banks of the array to sense the collector temperature rise. Thermocouples are also mounted in the same vicinity to indicate the air temperatures in each of the ducts. The air mass flow is derived by an array of pitot tubes manifolded together behind an air straightener in a manner to derive the average velocity head pressure sensed by a pressure transducer. Four thermocouples disposed 90 degrees apart sense the air temperature exiting from the velocity head pressure sensor for density compensation. A pyronometer senses total insolation in the plane of the collector. This information is fed into a signal conditioning unit for analog to digital conversion and integration of the insolation data. The information is recorded on magnetic tape in selected increments throughout a twenty-four hour day.

Additional thermocouples were added to sense temperatures in the locations indicated in the schematic of the test loop. The thermocouples mounted in the main inlet and exit ducts provide a check on the thermal performance of the overall 144 tube array. The thermocouples mounted in the manifolds in the inlet and exit air flow paths of the respective tube pairs sense the temperature rise in individual tube pairs. The temperature rise per tube pair is an excellent indication of air mass flow rate distribution. The thermocouples mounted in the annulus of the tubes are useful in sensing the annulus air temperature under both flow and no flow conditions. The latter sensed temperature may be useful in the logic used to control collector operation; the former sensed temperature provides a useful indication of the absorber surface temperature at its highest temperature axial position. Thirty of these temperatures require manual read out and recording; the remaining are available on a strip chart recorder. Static pressure taps sense the pressure levels in the inlet and exit air flow ducts. A list of the test equipment used in the air collector test loop is contained in Table 1.

B. Liquid collector array.

The standard SUNPAKTM module consists of 24 individual collector tubes manifolded together as shown in Figure 4. The module occupies a nominal 4' X 8' space when assembled. A typical module weighs about 110 pounds and will contain about 9 gallons of fluid when filled. The resulting load of the collector assembly is approximately 41b/ft² dry and 71b/ft² water filled. The effective collector area of a standard SUNPAKTM module is 27.4 square feet. It is this area that is used as the basis for describing collector performance and in price quotations.

The glass components are made with Owens-Illinois' KG-33 borosilicate glass to provide strength and thermal shock resistance. The fluid passageways inside the manifold are copper. High temperature silicone rubber O-rings and grommets are used for seals. The copper cup assembly and internal headers are encased in a molded urethane foam which serves as an insulating support structure. The materials have been chosen to resist damage to the collector by the high temperatures encountered in an unfilled collector exposed to the sun.

A complete parts list for the collector is shown in Table 2.

Water is the preferred heat transfer fluid due to its low cost and good thermal performance. However, other fluids such as glycol solutions are also acceptable. The low loss property of the SUNPAK™ collector makes use of water practical even in cold climates.

The SUNPAK™ manifold is designed to deliver water in a series flow pattern to its 24 tubes. This is accomplished with the use of 5/16 inch O.D. feeder tubes which channel water from and to the closed end of each collector tube. Figures 5A and 5B illustrate the flow pattern.

A plot of water flow rate versus pressure drop through a standard module is shown in Figure 6. A flow rate of 0.3 gpm/module is a typical operating rate and could be expected to give a temperature rise per pass of 20-30°F on a clear day. A minimum collector fill rate of 0.3 gpm/module is recommended. Collector fill flow rates greater than 0.3 gpm/module up to 0.5 gpm/module are suggested for optimum filling conditions. This range of flow will prevent two-phase flow in the feeder tubes which could result in entrapment of air during the filling procedure. Operation of a filled collector may be at any flow rate provided the recommended operating

pressure is not exceeded. Operation at flow rates above 0.2 gpm/module is recommended for best performance.

The collector has been designed to allow easy installation. Each component can be lifted by one man. Heavy lifting equipment is not necessary as long as there is sufficient access to allow the boxes containing the collector components to be carried to the mounting surface. After the end brackets and manifolds have been mounted on a surface, the tubes are simply inserted into their O-ring seals. The end bracket uses a plastic cap containing an adjusting screw to retain the tube in position when the hydraulic pressure of the system is applied to the tube. If a tube should need to be replaced, loosening the adjusting screw and a quick twist of the retaining cap frees the tube so that it can be pulled from its O-ring seal and a new tube installed. The simplicity of this system allows a 500 square foot array to be installed with approximately five man days of effort.

The manifold is designed with internal nominal 1 inch I.D. copper header pipes. Adjacent manifolds are coupled by a specially designed mechanical coupling provided with the collector. Additional mechanical couplings are available to connect a row of manifolds to piping to and from the array. As many as 15 modules may be joined together to operate from the internal headers. Longer arrays may be assembled using the internal headers, but attention should be given to the resulting header pressure drops and flow arrangement to prevent maldistribution of flow between the modules. For best distribution of flow, the inlet and outlet for a bank of modules should be at opposite ends of the array. Expansion of header pipes within each manifold is taken up by the mechanical coupling.

The collector is designed to mount on a tilted support surface provided by the customer. This could be a sloping roof or a sawtooth

structure on a horizontal roof. The surface should be flat. This means the manifold and end brackets of a single module may deviate from flatness by as much as 1/4". Pronounced surface irregularities, particularly along the length of the manifold may require the use of shims to obtain a smooth mounting surface.

When the collector is mounted on a watertight surface, a commonly employed technique to minimize the penetrations in the roof is to mount horizontal members on the roof to which the collector end brackets and manifolds can be attached. These horizontal members might be treated 2" X 6" lumber or galvanized metal channels which are spaced up off the roof with blocking to allow water drainage. The blocking should be flashed or caulked for watertightness.

The collector uses the mounting surface as a diffuse reflector. For best performance, the surface should have a non-glossy diffuse reflection characteristic such as would be obtained with a flat white paint. A surface which tends to be self-cleaning with rain water would be most desirable.

1. The steps for installation of the collector are:
 - a) layout the necessary dimensions on the mounting surface;
 - b) mount the end brackets, manifold brackets and tie rod assemblies;
 - c) mount the manifolds on the brackets, bolt in place and connect the header pipes via the mechanical coupling;
 - d) install insulating end caps and joint covers;
 - e) install the feeder tubes and collector tubes;
 - f) connect external piping and leak check.

The detailed description of each of these steps follows:

A single module will occupy a space of 4 feet wide and 8 feet tall. Provision should be allowed for inlet and outlet piping at the end of an array of modules and for the insulating foam end caps which project about six inches beyond the end of a module. Provision should also be made for removal of tubes during servicing which will require four inches of clearance at the ends of the eight foot dimension. Also, if the array consists of several horizontal rows of modules mounted directly above each other on an inclined surface, some provision should be made to allow access to any point in the array after the complete array has been installed.

After the positioning of the arrays has been determined, the surface should be marked with chalklines to locate the horizontal centerline of the manifolds and to position the end brackets. Chalklines at $46\frac{1}{4}$ inches ± 0 , $\pm \frac{1}{8}$ inches above and below the centerline can be used to position the end brackets as described in the next section. A line perpendicular to the horizontal lines will establish the starting positions for the manifold brackets and the end brackets. Then, four foot spacing can be marked off on the centerlines to establish the position of each module in the collector bank.

The manifolds are mounted with the manifold locator brackets, SK-2046. For proper positioning of the manifolds on the centerline, the brackets should be mounted with the top vertical surface on the centerline as shown in Figure 6. These brackets should be spaced 16 inches apart taking special care not to allow an accumulated error in positioning the brackets. The manifold brackets for use at the ends of each bank have been shortened to prevent interference with the insulating end cap. If the collector is mounted directly on a roof or other watertight surface,

appropriate steps should be taken to maintain the integrity of the roof.

The aluminum end brackets are mounted as shown in Figures 7 and 8 with the face of the channel flange at $44\frac{1}{8}$ " from the centerline of the manifold. Careful horizontal positioning of the end brackets is required to assure that the tubes will be held perpendicular to the manifold. Each module should occupy a four foot horizontal dimension.

The vertical positioning of the end brackets is done with the horizontal chalklines $46\frac{1}{4}" + 0. - \frac{1}{8}"$ from the manifold centerline which will serve as the centerline of the attachment hole in the bottom of the Z-shaped end bracket foot. See Figure 7. When the foot is attached to the mounting surface, the black rubber pad should be placed under the foot. At the end of each array of modules, the support foot is repositioned to interface with the hole drilled in the aluminum channel as shown in Figure 8. Also, at the end of each array of end brackets a spacer is provided for use between the channel and foot to maintain the proper end bracket height.

The tie rod brackets are mounted on the manifold centerline eight inches in from each end of the four foot horizontal dimension to match up with the $\frac{3}{8}$ " holes in the end bracket. A third tie rod is mounted in the center of the module or 24" from either end. This likewise, matches up with a hole in the end bracket. The tie rods are installed with $\frac{1}{4}$ - 28 nut with locknut at the center clip and $\frac{1}{4}$ - 28 nut and rubber faced washer at the aluminum end bracket. The tie rods should be adjusted so that the inside dimension between flanges of a module's end brackets is $88 - \frac{1}{4}"$ to $88 - \frac{1}{2}"$ when outward force is applied to the end brackets.

Before mounting, the manifolds should be checked to be sure that all O-rings, grommets and tube end seals have been installed and that the manifolds are free of foreign matter.

The location of the manifold on a surface is important for proper fluid distribution of the internal piping. The manifold must be positioned with the open section of the foot facing downward. A check on this may be made by noting that the manifold support bracket bolt fits into this opening. Reversing this location would create interference with the support bracket bolt.

Manifolds should be positioned on the support brackets one at a time. The mechanical coupling shown in Figure 9 should be located on each header line to connect the adjacent module. Once the manifold is properly located on its support, it should be tightened in place. The adjacent manifold coupling and manifold can then be connected in a similar fashion. Careful attention should be paid to horizontal positioning of the manifolds. Location of each manifold provides for a space of about 1/8" between the header pipes. This space, located within the mechanical coupling, provides for expansion of the assembly.

Soldering is not necessary within a row of modules. The piping to a row may be soldered to the manifold header lines. Care should be exercised in this operation so that internal brazing connections of the manifold are not damaged by excessive temperatures. An electric resistance soldering tool is recommended. Use of such a tool rather than a torch prevents scorching of the insulation materials. If a torch must be used, shields can be made to protect the insulation from the hot flame. Solder consisting of 95% tin and 5% antimony is recommended for attaching to the manifold header pipes.

The feeder tubes are in two parts to permit installation in confined places. The longer feeder tube (~47 inches) is installed in the manifold through the grommet from above. It should be inserted so that 43 inches are protruding above the grommet located on the centerline of the module. The shorter feeder tube is connected to this tube by a rubber connector. The feeder tubes should be installed with the square cut end in the rubber connector. The other end of each feeder tube should have a rubber guard as provided. This rubber guard protects the glass within the system from glass abrasion.

In confined spaces where the collector tube cannot be maneuvered through the end bracket and over the feeder tube, the two must be installed together. This is done by placing the feeder tube inside the collector tube when inserting the two elements from within the collector area through the end bracket opening. After clearance over the manifold, the feeder tube is first inserted through the grommet and the collector tube is located in its O-ring seat. The normal installation, with access to the collector, would install each feeder tube followed by the collector moving through the end bracket openings in the direction of the manifold. The positive locking end cup is installed over the collector tube with a twist as shown in Figure 10. After twisting the end cup into place, the adjusting screw is tightened finger tight only to bring the cup insert into contact with the collector tube. This will hold the tube in place when the system is pressurized. After this adjustment, the locknut should be secured. The collector tubes should be installed progressively with one on each side of the manifold. This will maintain the manifold perpendicular to the mounting surface rather than tilting due to uneven loading.

During the installation of the glass, the insulating joining covers should be placed between the tubes at manifold joints so they can be easily attached to the manifolds via a spring clip after pressure checking the system.

The piping connections to the collector may be made at either end of a bank. The top header (i.e., the pipe located furthest from the mounting surface) is the outlet header. The bottom pipe is the inlet header.

The connecting piping to each row of manifolds should be properly supported to prevent undue stress on the collector system. Expansion of external piping from the collector should be considered in this area. The headers within the collector manifold are compensated for expansion by the mechanical coupling. Support to the manifolds is not designed to cover the stresses that may be introduced by the connecting piping.

Vent valves near the inlet and outlet connections are recommended for several purposes. These parts can be used as air vents when the system is filled or drained. These valves may be manual or automatic depending on desired operation conditions. In an emergency no flow condition, the steam may be vented through these valves to protect the system from undue thermal and pressure conditions.

The maximum recommended operating pressure of the solar collector row is 30 psig. The recommended design is to provide for a pressure relief valve of 30 psig or less in the outlet header line to vent the collector in an emergency condition. The inlet of the collector should be maintained below 30 psig. This can be accomplished with a pressure regulator in the system. Each pressure relief valve should be vented properly to insure that steam and water are diverted safely. A pressure

relief valve should be provided to each row of manifolds. For multi-manifolded rows, each row which can be isolated from the system must have a safety relief valve.

The collector row should be checked for leaks at the coupling between modules and at the connecting piping. Next, the O-ring seal should be checked for leakage. Leak testing can be with either air or water. Water is the preferred method and can be used by pressurization of the system not to exceed 30 psig. In some systems or situations it may be desirable to use air to check for leaks. In these cases pressurization with low pressure air (~5 psi) and a soap solution is a convenient way to find leaks before a system is water filled. The collector should not be pressurized over 10 psi with air due to the potential hazard of flying glass if a tube would be broken. Note that air testing is not recommended during a bright clear day. Evenings or nights are suggested to reduce pressure-volume changes of air as it is heated in a closed system.

After leak checking, the insulating foam joint covers should be installed. The spring-clip device holds these covers together and permits access to this location for system servicing. The end caps can be shaped as necessary to make provision for the connecting piping. The cap should fit as closely as possible to the connecting piping and the manifold to reduce the possibility of heat loss. The caps are then glued into place by a waterproof mastic, such as white RTV silicone rubber, to complete the assembly.

The internal parts of the SUNPAK™ solar collector will approach temperatures in excess of 500°F while standing in bright sunlight. While the SUNPAK™ collector has been constructed with low expansion glass, filling the collector during mid-day portions of a bright day are not

recommended. Filling a collector in a bright sun could cause damage due to thermal shock. Introduction of a fluid into a hot tube could also result in the initial slug of fluid leaving the collector being a mixture of hot water and steam. The outlet from the collector on initial fill should be properly vented. The recommended procedure to avoid steam generation and potential thermal shock of the equipment is to fill in the early morning (before 9:00 a.m.) so that a high stagnation temperature is not reached.

A minimum fill rate of 0.3 gpm per module and typically 0.4 to 0.5 gpm per module is recommended to assure that air locks do not occur in the array. For example, a bank of 6 modules should be filled at a rate of 1.8 gpm. Filling in this manner will cause all 6 modules flowing in parallel with each other to fill uniformly. Filling at rates below the recommended minimum allows the possibility for two-phase flow in the feeder tubes and thus does not properly purge each module of air. Air entrapment can cause excessive back-pressure due to unbalanced columns of water in each module. This can result in one or more modules operating in parallel off the internal header pipes not to flow if the back pressure exceeds the pressure drop offered by neighboring modules.

Air locks may also be encountered when a partially filled array is refilled or whenever air is introduced into a filled array such as when the piping is drained for repairs. The piping system should be designed to prevent the introduction of air into the array during normal operation.

IMPORTANT

The boiling out of a collector as a means of emptying the collector for shut-down is not recommended. Under extreme insolation conditions, the collector could be damaged by thermal shock.

The very low loss coefficient of the SUNPAK™ collector affords it excellent freeze protection. The collector will gain enough energy even on the cloudiest of days to carry it through a below freezing night. It is recommended that all systems to the collector be properly insulated to avoid the problem of freezing a line to the collector and in turn isolating the collector in extreme cold conditions. Temperature monitoring of the collector fluid is suggested and heat may be added at night to keep the solar loop from freezing. A frequent practice for operation in weather below freezing requires that a pulse of flow be charged to the array once every one to four hours. This pulse need only be about one gallon per module to redistribute the contents of the array. Operation in this mode helps prevent freezing of piping leading to the array and will allow monitoring of the water temperature leaving the array. To prevent unnecessary loss from storage, a three way valve may be used to cause the recirculating water to by-pass the storage tank. In cold climates, where piping is normally traced with electrical tapes to avoid freezing, it is recommended that such tapes also be used in the solar loop.

The collector's tubular design tends to shed snow easily. Experience in Toledo has shown that even a nine inch snow storm did not cover the array. However, if an array should become completely covered by a

heavy snowfall, such that no insolation could reach the collector, there may be a danger of freezing the entire array. To prevent this, the entire volume of water in the collector should be exchanged with warm water at least once a day.

The liquid collector test loop is shown schematically in Figure 11. The test stand consists of two principal circulation loops, the collector loop and the load loop. The load loop is a water filled system including a storage tank. During normal operation, heat is removed from the collector by water circulation. This heat is transferred to the load loop through a liquid-to-liquid shell and tube heat exchanger. The energy is then added to storage or dumped by means of a water/forced air heat exchanger depending on the storage tank temperature. The test stand was designed with a sufficient range of temperature control to allow collector performance testing at various collector fluid inlet temperatures. Separate flow controls are used to control the amount of liquid entering each of the two banks (three modules each) of the array. Thus, the top and bottom banks may be tested independently for performance. A list of the major test stand components is given in Table 3.

The temperature of the collector loop is controlled by a 0-300°F pneumatic controller. The combined collector return fluid temperature is sensed and compared to the set point. When the outlet temperature of the collector is below the controller set point, the heat transfer fluid will bypass the heat exchanger. When the outlet temperature is within the controller proportional band width, the flow is proportionally introduced through the liquid/liquid heat exchanger. The load loop can be maintained

at temperature by adding electrical energy to the 82 gallon water storage tank. A temperature controller allows the setting of this minimum temperature.

The principal variables recorded for performance evaluation were T_{in} , T_{out} , T_{amb} , flow rate and total insolation in the plane of the collector. The temperatures were measured with type T thermocouples. Matched pairs were used for each bank to minimize the error in T . The thermocouples, manufactured by Omega Engineering, were encased in 1/4" OD stainless steel sheaths and installed directly in the fluid stream through compression fittings. The thermocouples were calibrated by the O-I Instrument Department with the standards used traceable to the National Bureau of Standards.

The flows to each collector bank were monitored visually with precision rotometers and regulated with needle valves. Turbine meters measure the flow rate in each bank and electronically input the measured value into the data logger.

The total insolation was measured with an Epply Model 8-48 black and white pyranometer mounted in the plane of the collector. The signal from the pyranometer was recorded as an instantaneous millivolt value at the time of each scan of the data logger. In addition, the millivolt signal was integrated between scans and an average insolation was recorded for the period between scans.

The data were recorded with a Fluke Summa II data logger on both printed paper and on magnetic tape. The data logger scanned each channel at regular pre-selected intervals. Intervals of 5 minutes were used for the data taken. Performance results were obtained from computer reduction of magnetic tape data.

C. Hybrid collector.

The details of design of the hybrid collector are considered to be proprietary since the system has not yet been offered for commercial sale. The collector tube elements as shown in Figure 12 are essentially identical to those used in the air and liquid collector arrays. The modular form as indicated in Figure 13 is similar to the liquid system. The module is 4' X 8' and contains 24 collector tubes, two steel serpentine, a manifold with internal header pipes, two end brackets and aluminum support structure with mounting clips. The hybrid collector removes heat from the collector tube in an indirect fashion. This is accomplished by means of a 3/8" OD steel tube bent into a serpentine configuration such that a U-shaped portion of the serpentine is placed inside each collector tube. Rather than transferring heat from the warmed absorber tube to the liquid or air directly, the heat is transferred from the absorber tube to the steel U-tube by radiation, conduction and convection through the stagnant air gap. Finally the heat transfers to the liquid flowing in the steel serpentine.

The advantages of this technique for heat transfer are many: The heat transfer fluid is enclosed in a steel piping system and does not depend on the integrity of the glass for containment. This means that glass breakage as a result of, for example, vandalism does not result in a release of fluid. The number of liquid seals is minimized and the two per serpentine that are required are flared-tube compression fittings which have high reliability. The closed steel piping system does not limit operating pressure. This approach to heat transfer results in a low thermal inertia of the collector allowing fast thermal response in the dynamic operation of the collector.

The test loop used for the hybrid collector was identical to that used and previously described for the liquid collector. A photograph composite of the three collectors is shown in Figure 14.

III. Description of the materials and method of manufacture of the manifold of the air collector array.

The air manifold consists of an inner and outer skin of fiber glass separated by an average of 1.5 inches. The volume between the skins is filled with a polyurethane material which is a three-part system. The three elements are mixed in liquid form and poured into the void volume. Depending on the amount or percent of catalyst used, foaming action will initiate in a matter of seconds and be essentially complete within a few minutes. The metal cups used for bottoming individual collector tube elements and for the containment of the O ring seal are mounted in place prior to the foaming process. The foam surrounds and binds to the metal parts during the foaming process resulting in an integral structure. The manifold is formed in two halves. The metal divider strip and seals are inserted between the two halves which are then mechanically joined by a suitable procedure.

The manifold fabrication procedure appeared to be simple in concept but proved to be quite difficult in practice. The fiber glass skins were fabricated by hand layup on wooden molds. Any skilled fiber glass boat manufacturer could handle these parts for prototype quantities. For convenience, the metal cups of the liquid system, as depicted in Figure 5 were modified by removal of the bottom half of the cup. The modified cups were silver soldered to a metal strip and then riveted in place individually to circumvent the problem of alignment which obtains due to tolerance stack-up problems. It was found that O rings at 15 to 20% stretch

provided a suitable seal to prevent the escape of a significant amount of polyurethane material during the foaming process.

A major problem in the manifold construction proved to be the foaming process. The three components of the isocyanurate foam were provided by the Mobay Laboratories, Pittsburgh, Pennsylvania. The materials were designated E643, Resin #249934 and catalyst C. The manifold volume to be filled was rather large at 2.8 ft.³ and the relatively thin cross sections and tortuous path the foam was required to follow complicated the process. At a target of 3.5 lbs/ft.² after completion of the foaming process, approximately 10 pounds of mixed material was required. The ratios used after considerable experimentation were .62 parts of E643 to .28 parts #249934 and .002 parts catalyst C. The catalyst was thoroughly premixed with the base material #249934 to assure good distribution of the catalyst in the final mixture since the catalyst did not react with the base material.

Positive displacement piston pumps were used to meter the flow rates of the two materials. The strokes of the two pumping systems were carefully calibrated to assure proper proportioning of the fluids. The fluids were injected at high pressure separately into a mixing head and the materials then thoroughly mixed in a high speed blender. With the catalyst ratio used, there was about 60 seconds from the time blending was initiated until foaming action started.

The volume ratio of the blended material as a liquid to that as a foam was approximately 50:1. As a result, the unit pressure levels reached during foaming were quite high at approximately 3 to 4 psi. With a surface area of about 16 ft.², some four tons of separating load had to be opposed.

A restraint system using 2" x 10" boards and the male mold used in laying up the inner skin was constructed. After assembly of all of the detail parts into a manifold half, the restraint system was applied and the entire assembly banded together. The liquid mixture was injected at one point centrally located in the manifold volume. Complete fill of the cavity was not attained in any of the four manifold halves; however, it proved to be a simple matter to pour a small amount of the premixed liquid material into the void areas to properly fill them. Some clean-up afterward was also required where the partially foamed material found escape holes and overflowed under the high foaming pressure. Had suitable equipment been available, the prefoam and probe injection process would have been used. In this process the volume ratios of the prefoamed material as injected and in its final state is of the order to 1:4. Foaming pressures are greatly reduced simplifying the restraint problem and the sealing to prevent the leakage of material.

Upon completion of the foaming of the four halves of the two manifolds, the metal divider strip was added along with gasket materials and the manifold halves mechanically secured together. The joint was sealed using an RTV silicone compound. The resulting manifold structurally was quite sound and also leak-tight. The sketches and drawings used as reference in the construction of the manifolds are attached.

IV. Complete description of the fabrication and assembly of the air collector array.

The air collector array consists of 144 collector tube elements, two manifolds each 12' long, tube end support brackets and a diffuse light reflecting background. The array was mounted on a 45° sloping roof which

completed the enclosure of a shed housing ducting, air handling equipment and instrumentation. Stringers of 2" x 10" board were attached to the roof to locate the manifolds and to support the light reflecting backing screen. The main body of the manifolds were inserted into gaps left in the 2" x 10" stringers. The diffuse light reflecting backing screen was 3/4" marine plywood painted with Dutch Boy flat white paint. The end support brackets were obsolete units used in the first generation liquid collector system. They consist of L-shaped sheet metal strips with holes punched on four inch centers. Plastic cups and lightweight springs locate the closed tube ends and support them in the sheet metal brackets. The brackets were mounted in place and the feeder tubes inserted into grommets captured in the metal divider strip in each manifold. The collector tube elements were inserted into the O ring seats in the manifolds and located and held in place by the spring end cap assemblies.

The roof penetrations for the air ducts were enclosed in sheet metal shaped to accommodate the access ports in the manifolds. The joints were sealed with an RTV silicone compound. The finished array is shown in Figures 1 and 2.

V. Data and formulation utilized in designing and constructing the air collector array.

The design of a multi-tubular collector array, using air as the heat transfer fluid, must provide simplicity in the attachment of the many discrete elements to an air manifold. The ease of the insertion and support of the tubes, the avoidance of air leakage at each interface, the uniformity of air flow distribution to each of the elements, a low thermal loss, aesthetics and cost are among the factors requiring consideration.

The thermal performance of evacuated tubular collector elements using a variety of heat transfer fluid characteristics is discussed by Beekley, et al.⁽¹⁾ Based on the Beekley analysis, it was decided to investigate a range of collector air flow rates from 4 SCFM to 8 SCFM per tube pair (8 to 16 pounds per hour per square foot). The resulting data would offer a flexibility in system design considering collector performance, manifold and ducting size and air pumping power.

Preliminary testing and analysis had characterized the air flow-pressure drop properties of the tubular collector element. While affected by dimensional values such as the diameter and location of the distributor tube relative to the absorber tube, a relationship of the form

$$\Delta P = \frac{CFM}{C} \quad \text{in. W.G.} \quad (1)$$

was established; for the particular system chosen a relation of

$$\Delta P = \frac{CFM^{1.76}}{23} \quad \text{in. W.G.} \quad (2)$$

was determined to be valid for a flow path where two collector tubes were operated in a series flow arrangement. The air flow path for the two tube series flow arrangement is indicated schematically in Figure 15. Each tube pair is operated in parallel flow relative to the manifold arrangement. Some researchers such as Grunes⁽²⁾ and Hughes, et.al.⁽³⁾ report that a two to one ratio of flow distribution can be accepted without serious impact on the thermal performance of an air driven evacuated tubular collector array. An objective of the one hundred forty-four (144) tubular collector design was to establish a test system (manifold arrangement) which would minimize the number of penetrations required through a roof/facade and which would maintain better than a two to one ratio of flow maldistribution,

the latter towards the potential of increasing the size of a total array without the necessity of adding to the number of roof penetrations required.

The manifold design chosen is shown schematically in Figure 16. The side by side air flow channels in the manifold will be noted. The unobstructed flow cross sectional area in the inlet and exit air flow ducts is about forty (40) square inches. The manifold transition flow areas to and from the two tube flow path are in the extensions at the top of the manifold. This arrangement was chosen to result in a minimum of the manifold extending above the false background which acts as the diffuse reflector and hides the larger portion of the manifold simply for aesthetic purposes. A secondary advantage is that the distributor tubes are disassociated from the main air flow path. This provides a somewhat improved transition region between the main flow sections of the manifold and the tubes.

The distributor tube is about 1" in diameter. Aluminum thin wall tubing was chosen for expediency; glass tubing would likely be used in volume production. The tubing is held in place and sealed to prevent air leakage to the exit duct in the manifold by a gasket mounted in holes in the thin metal plate dividing the inlet and exit air flow paths. The two foamed manifold halves utilize gasket material to hold the metal plate in place and seal against air leakage. A metallic dividing plate was found to provide adequate thermal insulation between the two flow paths.

The manifold is center fed and in the test arrangement eighteen tube pairs extend to either side of the transition feed connecting the manifolds to the main air ducting located within the building enclosure. A two tier manifold installation was selected towards providing flexibility in the

arrangement of future commercial installations of the air collector array. The manifold construction utilized an inner and outer skin formed by hand layup of a glass-reinforced high temperature resin. The tubular O ring seat, as a matter of convenience, utilized components developed for the liquid tubular collector. The manifolds were assembled in halves which included the connectors for the individual tube elements. The cavity was filled with a low density-high temperature polyurethane compound. The liquid mixture was pumped into the cavity and allowed to foam in place to form a relatively rigid structure. The manifold fabrication process was chosen as one having the future potential for high volume-low cost production following the teachings of the automotive field. A similar procedure is currently employed in the fabrication of the manifolds for the liquid tubular collector.

The collector tubes elements were taken from finished goods inventory of the liquid evacuated tubular collectors which are in pilot production. A collector tube element is shown schematically in Figure 17. The absorber tube is about 45 inches long and extends at its open end beyond the cover tube to fit into the tube socket and O ring seal. A selective coating is applied to the external surface of the absorber tube prior to assembly into the cover tube. Stability between the cover and absorber tubes is provided by a spring arrangement at the closed ends of the tubes. The spring provides radial support while allowing freedom in the axial direction to accommodate differential expansion. Attached to the support spring is a barium getter which contributes to the extended life of the vacuum which is lower than 10^{-4} torr applied to the volume between the two tubes.

Upon completion of the installation, air flow vs. collector pressure drop and leakage flow tests were performed. The variable speed fan was adjusted to provide a static pressure relative to ambient equal to that existing under rated flow conditions. The exit duct was blocked during the tests. A leakage flow of approximately 3% of rated flow was indicated. The air flow rate-pressure drop relationship was then measured. A best fit line on a log-log plot of the data indicates that the relationship for the manifold and collector tube assembly can be expressed as:

$$\Delta P = \frac{CFM^{1.84}}{27} \text{ in. W.G.} \quad (3)$$

At a flow rate of 4 SCFM/tube pair, the pressure drops are found to be equal within the limits of experimental error for the two tube pair and the one hundred forty-four tube array and verify that the pressure drop due to air flow in the manifold is negligible. An excellent flow distribution between tube pairs can also be inferred from the similarity in temperature rise measured for tube pairs. A significant increase in the number of tube pairs manifolded per roof penetration can therefore be anticipated without an adverse affect on air flow distribution. The agreement in the flow-pressure drop relationships between the 144 tube array and the 2 tube experimental testing also implies that the leakage and mass flow rates are within 5% of the measured values for the two systems.

The large pressure drop, often expressed as a potential problem when using air as the heat transfer fluid, is avoided by the flow path and manifold arrangement which is employed. At a 0.5 in W.G. pressure drop and flow rate of 4 SCFM per tube pair, the theoretical air horsepower required to drive a 144 tube array is only 0.023 HP. Even at a fan efficiency of 20%, a 0.1 horsepower drive would accommodate the pumping power requirements

of the collector or require about one percent of the useful solar energy collected. Of course, system pressure drop factors for ducting, dampers, storage and similar elements must also be considered. On the average, the system air pumping power should be less than 5 percent of the total useful energy collected. The energy loss associated with the efficiency of the air fan is captured in the air stream as a temperature rise and, in fact, does not represent an energy penalty to the system.

The control of thermal losses was an important consideration in the design of the air collector array. The losses are comprised of the reradiated energy from the absorber surface, the manifold surface conduction losses and miscellaneous conduction losses from the absorber surface through the absorber tube support spring to the cover tube and from the open end of the absorber tube through the glass seal and axially along the cover tube.

The radiation loss term, U_{LC} , may be determined from prior knowledge of the typical emittance value of the selective absorber surface. This has been the subject of extensive investigation as a part of the development of the liquid system. A typical value for the emittance, ϵ , is 0.070. The radiation loss is determined by:

$$Q_L = \sigma \epsilon (T_C^4 - T_a^4) \quad (4)$$

$$U_{LC} = \sigma (T_C^2 + T_a^2) (T_C + T_a) \frac{(T_C - T_a)}{(T_C - T_a)} \quad (5)$$

The conventions of solar collector analysis linearize the loss term as indicated in equation (5) and equate T_C to T_{in} to a good approximation for the analysis of the loss coefficient, U_L . Typical operating temperature conditions for an air driven collector $T_{in} = 200^\circ\text{F}$ and $T_a = 70^\circ\text{F}$ have

been selected for the following linearized analysis.

$$U_{LC} = \sigma (T_{in}^2 + T_a^2) (T_{in} + T_a) \quad (6)$$

$$U_{LC} = .1713 \times 10^{-8} \times .070(660^2 + 530^2) (660 + 530)$$

$$U_{LC} = .10 \text{ BTU/Hr.Ft.}^2\text{°F}$$

The total loss coefficient U_L is related to the losses associated with the collector tube element radiation losses and manifold conduction/convection losses by:

$$Q_{\text{Misc.}} + Q_{\text{Man}} + Q_C = Q_T = \pi A_C U_L \Delta T$$

$$U_{LM} A_M \Delta T + U_{LM} A_M \Delta T + \pi A_C U_{LC} \Delta T = \pi A_C U_L \Delta T$$

$$U_{LM} = \frac{1}{\frac{1}{h_1} + \frac{\Delta}{K} + \frac{1}{h_o} \frac{A}{A_i}}$$

$$h_1 = \text{air film coefficient} = 2.0 \text{ BTU/Ft.}^2\text{Hr.°F}$$

$$\Delta = 1.5 \text{ inches}$$

$$K = .16 \text{ BTU/Ft.}^2\text{Hr.°F/in.}$$

$$h_o = 5 \text{ BTU/Ft.}^2\text{Hr.°F}$$

$$A_i = 40.8 \text{ Ft.}^2$$

$$A_o = 56 \text{ Ft.}^2$$

$$U_{LM} = \frac{1}{\frac{1}{2} + \frac{1.5}{.16} + \frac{1}{5} \times \frac{28}{20.4}}$$

$$U_{LM} = 0.10 \text{ BTU/Ft.}^2\text{Hr.°F}$$

$$U_{LM1} = .049 \text{ BTU/Ft.}^2\text{Hr.}^\circ\text{F}$$

$$A_{M1} = .25 \text{ Ft.}^2$$

$$A_C = 35 \text{ Ft.}^2$$

$$U_L = \frac{U_{LM1} A_{M1} + U_{LM} A_M + \pi A_C U_{LC}}{\pi A_C}$$

$$U_L = \frac{.049 \times .25 + .10 \times 40.8 + \pi \times 35 \times .1}{\pi \times 35}$$

$$U_L = 0.14$$

The target performance contained in the contract requires a value of U_L equal to or less than:

$$\text{slope} = F_R \frac{A_C}{A_p} \pi U_L$$

$$F_R = \frac{1}{1 + \frac{\pi U_L A_C}{2\dot{m}C_p}}$$

$$U_L = \frac{\text{slope} \left(\frac{A_p}{A_C} \right) \times 2\dot{m}C_p}{\pi [2\dot{m}C_p - \text{slope } A_p]}$$

$$A_p = 82.8 \text{ Ft.}^2$$

$$\text{slope} = .39$$

The design air flow is 7.83 pounds per hour per square foot of collector area or 648 pounds per hour per module.

$$U_L = \frac{.39 \times 2.37 \times 2 \times 648 \times .241}{\pi (2 \times 648 \times .241 - .39 \times 82.8)}$$

$$U_L = 0.36 \text{ BTU/Ft.}^2\text{Hr.}^\circ\text{F}$$

Thus, the manifold design appears to be satisfactory from a heat loss standpoint.

The flow factor, F_R , is determined by the loss coefficient and the mass flow rate of the heat transfer fluid. At the design air flow rate of 7.83 pounds per hour per square foot and the design loss coefficient of 0.14 BTU/Ft.²Hr.°F, the value of F_R is:

$$F_R = \frac{1}{1 + \frac{\pi \times .14 \times 35}{2 \times 648 \times .241}} = 0.95 \quad (7)$$

Item 1,a of Article 1 Statement of Work states that the air collector shall have the potential of attaining an exit fluid temperature in excess of 250°F at an efficiency of greater than 40%. Previous experimental work has determined that the absorptance of the selective surface, α , is about 0.85. The properties of the borosilicate glass establish that the transmittance, τ , of the cover tube is 0.92. Analytical and experimental studies indicate that the light enhancement factor resulting from the spaced tubular collector and a diffuse light backing screen having a reflectance of 0.85 is 1.63. A reasonable value of the solar noon total insolation in the plane of the collector is 300 BTU/Hr.Ft.² and for ambient temperature on a day requiring space cooling, 80°F.

$$T_{in} = T_o - \Delta T \quad (8)$$

At an air flow rate of 7.83 pounds per hour per square foot and with the 300 BTU/Hr.Ft.² level of insolation a ΔT of at least 60°F would be expected.

$$\eta = \frac{A_c}{A_p} F_R \left[\frac{\tau \alpha S_{eff}}{S_{TP}} - \pi U_L \frac{(T_{in} - T_a)}{S_{TP}} \right] \quad (9)$$

$$\eta = \frac{35}{82.8} \times .95 \left[(.92 \times .85 \times 1.63 - \frac{\pi \times .14 (250 - 60 - 80)}{300}) \right]$$

$$\eta = 0.45 \quad (10)$$

Thus, the design of the air collector array would appear to exceed the requirements for the thermal performance of the array.

An important element is the overall design of the air collector system is air pumping power. Under the operating design conditions, the volume flow to a 72 tube module is 144 CFM. The pressure drop through the collector, as determined experimentally, is approximately 0.5 inches water gage. Using the standard equation from Mark's handbook, the air pumping power is:

$$\begin{aligned} \text{HP} &= 157.5 \times 10^{-6} pQ & (11) \\ &= 157.5 \times 10^{-6} \times 144 \times .5 \\ &= .011 \text{ HP} \end{aligned}$$

Assuming a blower efficiency of 20%, a .055 HP motor would supply the needs of the collector.

$$.055 \text{ HP} \times .746 \frac{\text{KW}}{\text{HP}} \times \frac{10^4}{2.93} = \text{BTU/Hr} \quad (12)$$

$$\text{Blower Power} = 140 \text{ BTU/Hr.}$$

$$\text{Collector Energy Gain} = 300 \times .45 \times 82.8 = 11,178 \text{ BTU/Hr.}$$

$$\frac{\text{Blower Power}}{\text{Collector Energy Gain}} = \frac{140}{11,178} = .0125 = 1.25\%$$

This appears to be an acceptable ratio of blower fan power requirements to useful energy gain.

VI. Test data and analysis.

A. Computer analysis program.

1. All day performance.

The evacuated tubular collector configuration presents perhaps unique requirements in the analysis of data. These include:

- a. The air collector has a long time constant in the order of 25 minutes. Also, the test loop had to be relatively unsophisticated due to cost considerations. Thus, the analysis procedures were derived to accommodate variations in collector inlet temperature and small fluxuations in air mass flow rate. Further, climatic conditions are quite variable in Toledo, Ohio with clear day conditions a very rare occurance. The air system has a very short fluid dwell time in the collector (in the order of seconds). The thermal mass of the air collector is quite low at .225 BTU/°F Hr.
- b. The liquid system has a short time constant (in the order of seconds). A problem in dealing analytically with the liquid system is the relatively long fluid dwell time in the collector (in the order of 20 minutes at a fluid flow rate of 10 lbs./Hr.Ft.²). Also, the collector has a rather high thermal mass at 3.225 BTU/Hr.°F. Cost restruictions in building the liquid test loop also resulted in a relatively crude control of collector fluid inlet temperature and mass flow rate. The highly variable solar insolation characteristic also applies to the liquid collector.

- c. The analytical model developed for the computer analysis of the performance of the collectors on an all day basis is based on accounting for all gain and loss terms over a small time increment. A schematic of the problem is shown in Figure 18 as is the equation for the collector efficiency for a given time increment. The relationship contains the usual $M C_p \Delta T$ term but in addition contains a term accounting for the change in thermal state of the collector during the time increment. The computer calculates the values for each increment of time during the test day starting one time increment after sunrise and stopping one time increment prior to sunset. The values of Q_u , Q_c and I_{TP} computed for each time increment are summed algebraically and the daily collector efficiency calculated as:

$$\eta_{day} = \frac{\sum Q_{u,i} + \sum Q_{c,i}}{A_p \sum I_{TP,c}} \quad (13)$$

2. The experimental data is captured in 5 minute increments throughout the day. The temperature and flow data are instantaneous values at the time of data recording. The insolation data is the total insolation integrated over the time increment. The temperature data for the air system is a differential value derived from a four element thermopile and a measure of inlet temperature in order to enter the tables to extract the temperature difference value. In the liquid system, the absolute temperatures at the inlet and the outlet are measured. The air mass flow is sensed following an air flow straightening device and a measure of total and static pressure from which the velocity head pressure can be derived. Average steam temperature is sensed by four thermocouples to provide the

necessary air density correction factor. The liquid mass flow rate is sensed by a turbine type flow meter. Ambient temperature is sensed by a shielded thermocouple.

B. Air Collector

1. Thermal performance.

The spaced evacuated tubular air collector, designed, built, tested and evaluated under the subject contract exceeded the predicted thermal performance significantly. This is shown graphically in Figure 19. The target was to demonstrate the capability of attaining an exit fluid temperature in excess of 240°F at a collection efficiency of greater than 40% on a daily operating basis. The data point is indicated by a circle. Perhaps of greater interest are the points indicated by the triangle and the square. These represent operating conditions of 240°F. Collector outlet air temperatures at 40% efficiency for insolation levels at solar noon in the order of 250 BTU/Hr.Ft.² and 200 BTU/Hr.Ft.² respectively. Note that the flat plate air collector would not even turn on under the latter condition.

In the northern climates relatively clear days under 0°F ambient conditions are often experienced. If it is assumed that a 160°F exit air temperature is necessary to supply the hot water or space heating requirements, the operating points designated by the X's are required for 40% efficiency at 305 BTU/Hr.Ft.² and 200 BTU/Hr.Ft.². The superiority of the evacuated tubular collector for this type of application is also evident. Thus,

the cost per unit of energy rather than the cost per square foot of collector should be considered for any particular application. Figure 20 presents a plot of the efficiency of the air collector based on integrated daily values at the design air flow of 8 lbs./Hr.Ft.². The trend for the efficiency of the collector to be better for the higher fraction diffuse radiation days will be noted. While the number of data points is not considered to be enough to quantify the results, Table 4 indicates collector parameter values derived for days with high values of diffuse radiation (1300 BTU/Ft.² day or less) versus days having higher values of clearness index. The trend is considered to be significant both in the magnitude and direction of the change in performance. An increase in the number of test points would allow the effect to be quantified. Since the difference is almost 11% at the intercept and becomes even greater as the $\Delta T/I$ values increase, collector sizing could be impacted significantly in areas of the country where a high cloudiness index is experienced.

The plots for data derived at air flow rates of approximately 8, 12 and 16 Lbs./Hr.Ft.² and a composite plot for each of the air flow rates are contained in Figure 21A through 21D. Table 5 contains a compilation of the important collector parameters for each air mass flow rate. The data appears to be rather consistent. A rigorous solution for each of the parameters based on experimental data is possible using relationships derived by Beekley⁽¹⁾. However, the derivation of U_L was based on heat transfer from the absorber surface to the air only by conduction. If this were the case, as the flow rate increases the value of U_L should decrease since increasing air flow rates cause

a decrease in the film coefficient. The radiation term between the inner glass wall of the absorber tube and the outer wall of the distributor tube would appear to be an important mechanism increasing in importance as the air flow rate is reduced. Further experimental testing and analysis is required to sort out the effects. Exploration of the radiation factor would require a probing of the air temperature in the annulus flow path as a function of axial position and perhaps the recording of the axial surface temperatures of distributor tube.

The importance of resolving the issue is evidenced by the large variation in the incident angle modifier as a function of air flow rate. Since this parameter is affected only by the optical characteristics of the collector, the value of the modifier would be expected to be a constant. A sensitivity analysis was performed to see if errors in the measurement of air flow rate (the most suspect term in most air experiments) would affect the results. Only a second order affect was determined. Thus, the definition of U_L , its determination experimentally and the resulting affect on the term, F_R , are considered to be of great importance. If it were found through further test and analysis that the F_R term turned out to be significantly lower at an air flow rate of 8 Lbs./Hr.Ft.², the value of the incident angle modifier would increase.

The basic equation for the evacuated tubular collector is:

$$\eta = \frac{A_c}{A_p} F_R \left[\phi \frac{\tau_d I_{eff.}}{I_{TP}} - \pi U_L \frac{(T_{in} - T_a)}{I_{TP}} \right] \quad (14)$$

The ratio, A_c/A_p , relates the absorber tube cross sectional area to the total effective absorption area of the collector. It acts as the modifying term to the level of radiation reaching the absorber surface and to

the loss coefficient, U_L . Studies by Beekley, et al(1), developed that the ratio of I_{eff}/I_{TP} at solar noon was approximately 1.6. The boundary conditions assumed that the cover tube caused the shadowing effect limiting the radiation reaching the backing screen. The absorber surface cross section was used in determining the view factor from the light strips on the backing screen to the absorber surfaces. Previously released data, Moan(2) and Moan(3) used the analytically derived value for the ratio of I_{eff}/I_{TP} in the analysis and presentation of experimental data relating to the evacuated tubular air collector.

Rewriting the equation:

$$\eta = F_R \left[\tau\alpha \left(\frac{A_c}{A_p} \phi \frac{I_{eff}}{I_{TP}} \right) - \frac{A_c}{A_p} \pi U_L \frac{(T_{in} - T_a)}{I_{TP}} \right] \quad (15)$$

For the collector configuration under test, $A_c/A_p = .42$.

$$\eta = F_R \left[\tau\alpha (\phi \times .42 \times 1.63) - .42 \pi U_h \frac{(T_{in} - T_a)}{I_{TP}} \right]$$

$$\eta = F_R \left[\tau\alpha (\phi \times .68) - 1.32 U_L \frac{(T_{in} - T_a)}{I_{TP}} \right] \quad (16)$$

Compared to the efficiency relationship of a flat plate collector:

$$\eta = F_R \left[K_{\alpha\tau} (\tau\alpha) - U_L \frac{(T_{in} - T_a)}{I_{TP}} \right] \quad (17)$$

It will be noted that the gain term for the spaced evacuated tubular collector with a diffuse radiation backing screen has been derated to less than unity. The radiation passing between the spaced tubes does not reach the absorber surface as effectively as with a full flat absorber surface. This is one major reason that the intercept values of the spaced evacuated tubular collector are, in many cases, lower than the intercept values reported for flat plate collectors.

2. Solar radiation enhancement.

Because of the cylindrical shape of the evacuated tubular collector, solar radiation enhancement features may be added easily and inexpensively. One simple approach is to simply space the tubes one from the other and add a plain diffuse reflecting screen behind the array. The thermal performance of evacuated tubular collector elements under a variety of spacing between tubes and between the planes of the tubes and the backing screen was investigated by Beekley, et. al.⁽¹⁾. It was predicted that a spacing of two cover tube diameters would produce the same amount of useful energy on a daily basis as would a close packed array with contiguous tubes. Thus, the cost effectiveness of each tube is almost doubled. The light enhancement factor for the spaced tubular collector is expressed as:

$$I_{eff} = I_{gp} \{ \text{See } \Omega + (1 + \rho \Delta) \left[1 + \frac{1 - \rho \Delta}{1 + \rho \Delta} (\text{See } \Omega - 1) \right] \}_{w=0} \quad (18)$$

Analysis predicts that:

$$\frac{I_{eff}}{I_{TP}} = 1.63_{w=0} \quad (19)$$

In similar manner, the loss term is increased through a combination of the area ratio, A_c/A_p and π to account for the cylindrical surface area of the absorber surface. It is, therefore, vital that the loss coefficient, U_L be kept as low as possible. The principal factor affecting the value of U_L is the emittance of the absorber surface. The need for relatively large air flow cross sections in the manifolds results in a significant surface area exposed to ambient conditions leading to conduction and convection loss terms for the collector array.

The flow factor, F_R , is defined by Beekley⁽¹⁾ as:

$$F_R = \frac{1}{1 + \frac{\pi U_L A_c}{2 \dot{m} C_p}} \quad (20)$$

and the loss coefficient, U_L , from the experimentally determined slope of the efficiency curve is:

$$\text{Slope} = -F_R \frac{A_c}{A_p} \pi U_L \quad (21)$$

It should be noted that in the derivation of F_R by Beekley⁽¹⁾, the U_L term does not contain a radiative factor between the inside surface of the absorber tube and the outside surface of the distributor tube. This is probably not important in a liquid system. In an air system, however, where the film coefficients result in relatively poor heat transfer from the inner glass surface to the air stream it could be of significance.

The data developed in Chart 5 neglects the radiative heat transfer mechanism. It is believed that the experimentally derived slope and intercept data are real. The inconsistency is the variation of the incident angle modifier with air flow since it is a function only of the optical properties of the collector array. Also, the dependence of collector performance on percent diffuse radiation is believed to be real. This suggests that the derivation of I_{eff} could be suspect principally because of the boundary conditions which were assumed in its derivation. It is beyond the scope of this final report to develop a more rigorous relationship. Additional time both for analysis and for the development of further test data to fill in gaps would be required.

One can test the impact of changes in the critical collector parameters towards the development of greater consistency in the experimental

data. Simulation studies by Eberlein⁽⁴⁾ suggests that the loss coefficient will remain essentially constant over the air flow range of interest.

$$\text{Slope} = \frac{Ac}{Ap} F_R \pi U_L \quad (22)$$

$$\eta_o = \frac{Ac}{Ap} F_R \tau \alpha \phi (I_{eff} / I_{tp}) \quad (23)$$

The experimental data was investigated using reasonable values for U_L and assuming that U_L remains essentially constant for the range of air mass flow rates used in deriving the experimental data. The results are shown in Table 6. This treatment of the data provides consistent values for the optical constants; viz., the incidenced angle modifier, ϕ , and the radiation enhancement term (I_{eff}/I_{tp}). The expressions relating F_R to the loss coefficient, U_L and air mass flow, m , (equations 20 and 21) appear to require revision. The radiative heat exchange between the inside glass surface of the absorber tube and the external surface of the distributor tube must be accounted for. An optical constant of 2.0 for the combination of the radiation enhancement term at solar noon times an incident angle modifier integrated over the day can be supported by analysis. The radiation enhancement term at solar noon would approach 1.87 using the absorber tube diameter as the shading boundary and the incidenced angle modifier integrated on a daily basis would approach 1.07. This would be satisfied by the experimental data selecting a loss coefficient, U_L , of 0.17 BTU/°F, Hr.Ft.².

3. Collector time constant.

The collector time constant was measured experimentally by two procedures. The first procedure allowed the collector to stagnate

under no flow conditions all morning on a clear day. Air flow at ambient conditions and at approximately 8 Lbs./Hr.Ft.^2 was introduced leaving the array uncovered. Temperatures in the annulus of two tubes, one located in the upper bank and one located in the lower bank, were recorded in five minute increments. The exit duct air temperature was also recorded. The results are presented in Figure 22.

In the second procedure, the collector was allowed to stagnate all morning under no flow conditions. The array was then covered using a black plastic shield and simultaneously air flow at ambient temperature and at a rate of approximately 8 Lbs./Hr.Ft.^2 was introduced. The temperature in the annulus of two tubes, one in each bank, was recorded in five minute increments. The experimental results are presented in Figure 23.

In both cases, the experiment was continued until a steady state temperature was reached. The time to reach 37% of the difference between the initial and final temperatures was approximately 25 minutes. This is a rather long time constant compared to that reported for flat plate collectors using a liquid as the heat transfer fluid. The long time constant caused a significant impact on the analysis of experimental data.

An attempt was made to apply the approach to the experimental evaluation of the incident angle modifier contained in ASHRAE 93-77. Five of the clearest test days experienced during the January to June 1977 test period were chosen. The collector was operated with ambient inlet air and full overboard dump. This resulted in the collector operating point being as near as possible to the η_0 intercept point. The

results of the analysis are shown in Figure 24. The time increment chosen was equal to the time constant or 25 minutes. The lines represent the linear least squares fit of data for five days and two 25 minute increments equally spaced around solar noon. The scatter in the data is evident. Yet the scatter in data using all day time increments per data point is quite acceptable. And in the latter case, it was not necessary to maintain close control on the collector fluid inlet temperature or to throw out days having intermittent cloudyness or high fractions of diffuse radiation conditions.

4. Collector sensitivity to diffuse radiation.

During the course of analysis on the experimental data, a degree of scatter in the data was noted. After a series of manipulation of the data, it was found that there was a distinct difference in the collector performance for operating days having 1300 BTU/Ft.^2 day of radiation or less compared to days having higher levels of radiation. The data points and linear least squares fit are shown in Figure 20. It is apparent that the collector is quite sensitive to the fraction of diffuse radiation. Unlike many collectors such as the flat plates and concentrating configurations, the level of collector performance increases as the fraction of diffuse radiation becomes greater. This could impose a limiting condition on the minimum radiation level allowed for the rating of collectors on an all day basis. There were an insufficient number of data points to investigate performance in incremental differences in the daily radiation level. Further testing and analysis could allow a quantification of this important characteristic.

C. Liquid collector.

The performance of the 144 tube liquid system, using water as the heat transfer fluid is shown in Figure 25. The performance of the air collector at the design mass flow rate of 8 Lbs./Hr.Ft.² has been added for ease of reference. There are two notable differences in the experimental data. The liquid system has a significantly lower intercept value and a much shallower slope than the air collector. However, the results are considered to be inconclusive. There are no data points for the liquid system near the intercept. A simple sensitivity analysis indicated that just three data points assuming an efficiency of .55 at a $\Delta T/I$ value of .05 would change both the intercept and slope values dramatically.

D. Stagnant air liquid (SAL) collector.

The performance of the 144 tube SAL system, using DOW Therm J as the heat transfer fluid, is shown in Figure 26. The performance of the air collector at the design mass flow rate of 8 Lbs./Hr.Ft.² has been added for ease of reference. As with the liquid system, the lack of experimental data near the intercept operating point prevents the drawing of significant conclusions.

LIST OF REFERENCES

- (1) D. C. Beekley and G. R. Mather, "Analysis and Experimental Tests of a High Performance, Evacuated Tubular Collector." Presented at the ISES Los Angeles, California, 1975.
- (2) K. L. Moan, "An Analysis of the Low Loss Evacuated Tubular Collector Using Air as the Heat Transfer Fluid." Presented at the ISES Orlando, Florida, 1977.
- (3) K. L. Moan, Update of (2), Presented at the R & D Branch Solar Division, ERDA Contractors Conference. Reston, Virginia, 1977.
- (4) M. B. Eberlein, "Analysis and Performance Predictions of Evacuated Tubular Solar Collectors Using Air as the Heat Transfer Fluid," Master's Thesis, University of Wisconsin, Madison, 1976.

Figure 2

FIGURE 1

ERDA Air Collector Test Loop

Figure 3

TABLE 1

List of Test Equipment
Air Collector Test Loop

<u>Measurement</u>	<u>Instrumentation</u>
I_{Tp} : Solar total radiation in plane of collector	Apply Precision Pyranometer with integrator.
T_a : Ambient Temperature	Copper-Constantan radiation shielded thermocouple
$T_{out} - T_{in}$: Collector bank temperature rise	Four element copper-constantan thermopiles
T_{in} : Inlet temperature to each collector bank	Copper-constantan thermocouples
m : Air flow rate to collector	Air Monitor Corp. Duct Air Monitor Device
ΔP : Collector Pressure Drop	Magnehelic, Model No. 2002C, Dwyer Instruments, Inc.
Data Logger:	John Fluke Co. (Model 2240 Data Logger)

Figure 4

Table 2
SUNPAK™ Parts List

<u>Number Required per Module</u>	<u>Part Number</u>	<u>Part Identification</u>
1	SK-3042	Manifold
4	SK-2046	Manifold Locator Bracket
12	SK-2048	Grommets (in Manifold)
24	SK-2051	O-Rings (in Manifold)
24	SK-2050	End Seals
12	SK-2049-1	Feeder Tubes
12	SK-2049-2	Feeder Tubes
12	SK-2397	Feeder Tube Connector
2	SK-2848	Tube End Support
2	SK-2870	Clip
2-4	SK-2880	Mounting Spacer
2-4	SK-2875	Mounting Pad
0-2	SK-2989	Shim Spacer
6	SK-3851	Support Rod
3	SK-3852	Center Bracket
24	SK-3048	Tube Support Cup Assembly
24	SK-2060	Collector Tube Assemblies
As Req'd	SK-3045	End Cap Foam
As Req'd	SK-3046	Insulating Series Connection
2	SK-3047	Tube Coupler

Figure 5

Figure 6

Figure 7

Figure 8

Drawing 9

Figure 10

FIGURE 11

1. Pump: Goulds Pump, Model 3642, 1/2 H.P.
2. Flowmeters: Fischer Porter, 10A3000 Series, Glass Tube indicating Flowmeters
1.12 GPM full scale (water)
3. Flowmeters: Fischer Porter, 10A3000 Series, Glass Tube indicating Flowmeters
9.5 GPM full scale (water)
4. Turbine Meters: Flow Technology Standard Line Turbine Flowmeter
#FT-6-8M5-LB, 0.3 to 3 gpm with silicone oil, 0.5 to 5.0 gpm with water
5. Solar Collector: Three module bank of stagnant air liquid collector (SAL)
6. Pneumatic Control Valve: Fisher Controls Co.
Type 513 Diaphragm Acuator with Design GS Body
7. Pneumatic Controller: Ametex Control Div.
Model 41KM4145DD3224
Range 0-300°F
8. Liquid/Liquid Heat Exchanger: American Standard
SSCF Heat Exchanger
Part 47M301-2A25
9. Air/Liquid Heat Exchanger: Modine Propeller Type Unit Heater
Model HS-160
24,600 Btu/hr at 2.5 GPM
200°F inlet water
180°F outlet water
60°F inlet air
10. Expansion Tank: MCS Expansion Tank
8 gallons, Model 3667K1
11. Needle Valve: Whitey Union Bonnet Regulating and shut off valve
Catalog Number - 8RF8
12. Pneumatic Controller: Ametex Control Div.
Model 41KM4145DD3224
Range 0-200°F
- Pneumatic Control Valve: Fisher Controls Co.
Type 513 Diaphragm acuator with Design GS Body
13. Pump: Bell and Gossett Booster Pump
Model no. (Series 100), 1/12 H.P.
14. Flowmeters: Fischer Porter
10A3000 series Glass Tube indicating Flowmeters
Range: 6.8 GPM (water)
15. Storage Tank: Crane
82 gal. Hot water tank

SUNPAK™ SAL Collector Tube

Figure 12

Figure 13

AIR COLLECTOR

LIQUID COLLECTOR
FIGURE 14

HYBRID COLLECTOR

**TWO BANK AIR COLLECTOR ARRAY
SERIES/PARALLEL AIR FLOW PATH
(36 TUBE PAIRS PER BANK)**

67

Figure 15

Figure 17

EVACUATED TUBULAR COLLECTOR: THERMAL BALANCE SCHEMATIC

DAILY PERFORMANCE INDEX:

$$\begin{aligned} \bar{\eta} &= \frac{\sum (\dot{m} C_p)_i (\bar{T} - \bar{T}_{i_n})_i + \sum \bar{\tau} \Delta t_i \overline{MC_p} (\bar{T} + \bar{T}_{i_n} - 2 T_0)_i}{A_p \sum \bar{S}_{pi}} \\ &= \frac{A_c}{A_p} \left[\alpha Z \frac{\sum \bar{S}_{eff i}}{\sum \bar{S}_{pi}} - \pi U_L \frac{\sum (\bar{T}_{i_n} - \bar{T}_a)_i}{\sum \bar{S}_{pi}} \right] \\ \bar{\tau} &\equiv \frac{1}{1 + \frac{U_L A_L}{2 \dot{m} C_p}} \end{aligned}$$

Figure 18

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

Figure 19

Air Driven Spaced Evacuated Tubular Collector Design $\dot{m} = 8 \text{ LBS/HR. FT.}^2$

FIGURE 20

Diffuse VS. Clear Day Performance
Flow Rate Equal 8 LBS/HR. FT.²

<u>I_{TP}</u>	<u>n</u>	<u>SLOPE</u>	<u>U_L</u>	<u>F_R</u>	<u>Φ</u>
< 1300	0.638	-0.175	0.135	0.977	1.21
> 1300	0.576	-0.193	0.149	0.974	1.10
ALL DAYS	0.581	-0.178	0.137	0.976	1.10

TABLE 4

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

FIGURE 21-A

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

FIGURE 21-B

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

FIGURE 21-C

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

FIGURE 21-D

<u>$\dot{m}$: LBS/HR.FT.²</u>	<u>η_o</u>	<u>SLOPE</u>	<u>U_L</u>	<u>F_R</u>	<u>ϕ = I.A.M.</u>
8	0.581	-0.178	0.137	0.976	1.10
12	0.624	-0.188	0.144	0.983	1.18
16	0.655	-0.196	0.150	0.987	1.23

$$\bar{\eta} = \frac{A_c}{A_p} F_R \left[\phi \tau \alpha \frac{I_{eff}}{I_{TP}} - \pi U_L \left(\frac{\sum T_{in} - \sum T_A}{\sum I_{TP}} \right) \right]$$

TABLE 5

Table 6

Investigation of Collector Paramaters
Loss Coefficient $U_L = X$

$U_L = .16$

M	η	Slope	U_L	F_R	$\phi \times (I_{eff}/I_{TP})$
8	.581	- .175	.16	.83	2.13
12	.624	- .188	.16	.89	2.13
16	.655	- .196	.16	.93	2.14

$U_L = .17$

8	.581	- .175	.17	.78	2.26
12	.624	- .188	.17	.84	2.26
16	.655	- .196	.17	.87	2.29

$U_L = .15$

8	.581	- .175	.15	.88	2.01
12	.624	- .188	.15	.95	2.00
16	.655	- .196	.15	.99	2.01

**COLLECTOR TIME CONSTANT TUBES UNCOVERED
AND STAGNATED FLOW AT 4SCFM INTRODUCED**

AMBIENT TEMP. = 84°F

□ INSULATION

• LOWER TUBE ANNULUS

× UPPER TUBE ANNULUS

⊙ EXIT DUCT AIR TEMP.

INLET DUCT AIR TEMP. $\hat{=}$ 90°F

Figure 23

AIR DRIVEN SPACED EVACUATED TUBULAR COLLECTOR

FIGURE 24

Figure 25

**SPACED EVACUATED TUBULAR COLLECTOR
CHARACTERISTIC DAILY THERMAL PERFORMANCE
STAGNANT AIR LIQUID COLLECTOR**

Figure 26

SECTION A-A FULL SIZE

MATERIAL - 18GA. (.046) LOW CARBON STEEL
PICKLED & OILED

DEC 13 1977

J				
H				
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E				
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B				
A				
CHG.	DESCRIPTION	BY	DATE	MF

OWENS-ILLINOIS, INC.
CORPORATE TECHNOLOGY
TOLEDO, OHIO

RAIL

SOLAR ENERGY

NUMBER
SK-2994
SHEET 1 OF 1 SIZE C

MARK PART NUMBER (X)

USED ON ASSEMBLY NO. SK-2993

TOLERANCE UNLESS OTHERWISE NOTED	2 PLACE DEC. ± .02	3 PLACE DEC. ± .005	DEFINITION OF SYMBOLS	MATERIAL -	PROJECT NUMBER
			⊥ PERPENDICULARITY		C-2331
			// PARALLELISM	HEAT TREAT	DES'D 11
			□ FLATNESS	CASE DEPTH	DR'N 716172
			◎ CONCENTRICITY	HARDNESS	CK'D 11
				SUR. TREAT	
				6-REQ'D	SCALE 1/4" = FULL
UNLESS SPECIFIED, MACHINED SURFACE ROUGHNESS NOT TO EXCEED	✓		RADII UNLESS OTHERWISE NOTED		

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C00/2919--3

A
B
C
D

2 3 4 5 6 7

MATERIAL - 3003 ALUM.

DEC 13 1977

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CHG.	DESCRIPTION	BY	DATE	MF
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OWENS-ILLINOIS, Inc.
CORPORATE TECHNOLOGY
TOLEDO, OHIO

DIVIDER-MANIFOLD

3003 ALUM. SYSTEM

NUMBER
36-2995

SHEET 1 OF 1 SIZE C

MARK PART NUMBER (X)

USED ON ASSEMBLY NO. 2991

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TOLERANCE UNLESS OTHERWISE NOTED	2 PLACE DEC. $\pm$	DEFINITION OF SYMBOLS	MATERIAL NOTED	PROJECT NUMBER		
	3 PLACE DEC. $\pm .005$			DES'D	11	
CORNERS AND/OR EDGES BROKEN	OUTSIDE RAD. MAX.	$\perp$ PERPENDICULARITY	HEAT TREAT	DR'N	7/11/76	Kiza
	INSIDE RAD. MAX.	// PARALLELISM		CK'D	11	
UNLESS SPECIFIED, MACHINED SURFACE ROUGHNESS NOT TO EXCEED	$1/25$	$\square$ FLATNESS	CASE DEPTH			
	$1/6$ RADII UNLESS OTHERWISE NOTED	$\odot$ CONCENTRICITY	HARDNESS			
			SUR. TREAT			
			1-REQ'D	SCALE	$1/8" = 1"$	

2 3 4 5 6 7

A

B

C

D

A

B

C

D

#	DIMENSION A	SK-NUM.	# REQ
1	72.75	3014	2
2	17.38	3015	2

MATERIAL
ALUMINUM 1100 .025 THICK SATIN FINISH
ON OUTSIDE SURFACE

DEC 13 1977

J				
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CHG. DESCRIPTION BY DATE MF

OWENS-ILLINOIS, INC.
CORPORATE TECHNOLOGY
TOLEDO, OHIO

TRIM STRIP

NUMBER
SK-3014, SK-3015

SHEET 1 OF 1

C
SIZE

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MARK PART NUMBER (X)

USED ON ASSEMBLY NO.

TOLERANCE UNLESS OTHERWISE NOTED	2 PLACE DEC. $\pm .010$	DEFINITION OF SYMBOLS $\perp$ PERPENDICULARITY $\parallel$ PARALLELISM $\square$ FLATNESS $\odot$ CONCENTRICITY	MATERIAL NOTED	PROJECT NUMBER C-2331	
	3 PLACE DEC. $\pm .005$			DES'D	11
CORNERS AND/OR EDGES BROKEN	ANGLES $\pm$	UNLESS SPECIFIED, MACHINED SURFACE ROUGHNESS NOT TO EXCEED	HEAT TREAT	DR'N	619176 L74
	OUTSIDE RAD. MAX.			CK'D	11
	INSIDE RAD. MAX.	$\frac{1}{4}$ RADII UNLESS OTHERWISE NOTED	CASE DEPTH		
			HARDNESS		
			SUR. TREAT		
			SCALE	DOUBLE	

2 3 4 5 6 7

MATERIAL - MAKE FROM SK-2038

DEC 13 1977

MARK PART NUMBER (X)

USED ON ASSEMBLY NO. SK-2993

TOLERANCE UNLESS OTHERWISE NOTED	2 PLACE DEC. $\pm .01$	DEFINITION OF SYMBOLS (Total Indicator Reading)	MATERIAL AS SHOWN	PROJECT NUMBER C-2331		Owens-Illinois, Inc. CNPD - DEVELOPMENT CENTER TOLEDO, OHIO AIR SYSTEM
	3 PLACE DEC. $\pm .005$			DES'D 11	DR'N 7/1/76 and	
CORNERS AND/OR EDGES BROKEN	OUTSIDE RAD. MAX.	CONCENTRICITY	HEAT TREAT	CK'D 11	CUP - COLLECTOR	NUMBER SK-2999
	INSIDE RAD. MAX.	ROUNDNESS	CASE DEPTH	SCALE 2 x SIZE		
UNLESS SPECIFIED, MACHINED SURFACE ROUGHNESS NOT TO EXCEED	RADIO UNLESS OTHERWISE NOTED	FLATNESS	HARDNESS	SHEET 1 OF 1	C	SIZE
		SYMMETRY	SUR TREAT 36 REQ'D			
		PARALLEL				
		PERPENDICULAR				

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2

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4

A

SOURCE

PHILPOTT RUBBER CO
20377 EAST 50TH ST.
CLEVELAND OHIO
44115
PART # 1A931-AIG-22

DEC 13 1977

MATERIAL - SILICONE RUBBER
50 DUROMETER. SHAPE "A"

B

MARK PART NUMBER (X)

USED ON ASSEMBLY NO

TOLERANCE UNLESS OTHERWISE NOTED	2 PLACE DEC. $\pm$ 0.010	DEFINITION OF SYMBOLS (Total Indicator Reading)	MATERIAL	50 DUROMETER	PROJECT NUMBER	C-2331	
	3 PLACE DEC. $\pm$ 0.005						
	ANGLES $\pm$						
CORNERS AND/OR EDGES BROKEN	OUTSIDE RAD. MAX.	CONCENTRICITY	HEAT TREAT	DES'D	11		
	INSIDE RAD. MAX.	ROUNDNESS	CASE DEPTH	DR'N	717.10	361	
UNLESS SPECIFIED, MACHINED SURFACE ROUGHNESS NOT TO EXCEED	RADI UNLESS OTHERWISE NOTED	FLATNESS	HARDNESS	CK'D	11		
		SYMMETRY	SUR. TREAT				
		PARALLELISM	SCALE	FULL			
		PERPENDICULARITY					

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CHG.	DESCRIPTION	BY	DATE	MF

OWENS-ILLINOIS, INC.
CORPORATE TECHNOLOGY
TOLEDO, OHIO

GROMMET

SQUARE ENERGY - AIR

NUMBER
SK-3013

SHEET 1 OF 1
SIZE A

2

3

4

MATERIAL -
GLASS REINFORCED EPOXY

MARK PART NUMBER			USED ON ASSEMBLY NO. <u>SK-2997</u>				CHG.	DESCRIPTION		BY	DATE	MF	
TOLERANCE UNLESS OTHERWISE NOTED	2 PLACE DEC. $\pm$ <u>0.010</u>		DEFINITION OF SYMBOLS (Total Indicator Reading)		MATERIAL <u>SEE DRAWING</u>		PROJECT NUMBER <u>C-2331</u>		OWENS-ILLINOIS, INC. CNPD - DEVELOPMENT CENTER TOLEDO, OHIO				
	3 PLACE DEC. $\pm$ <u>0.005</u>												
	ANGLES $\pm$			CONCENTRICITY	HEAT TREAT		DES'D	<u>1 1</u>	MANIFOLD COVER SOLAR ENERGY				
CORNERS AND/OR EDGES BROKEN	OUTSIDE RAD. MAX.			ROUNDNESS			CASE DEPTH	DR'N					<u>7/16 IN</u>
	INSIDE RAD. MAX.			FLATNESS	HARDNESS	CK'D	<u>1 1</u>	NUMBER		D
			SYMMETRY	SUR. TREAT	SCALE		<u>1/8 (4:1)</u>	SHEET		<u>1</u>	OF	<u>1</u>
	UNLESS SPECIFIED, MACHINED SURFACE ROUGHNESS NOT TO EXCEED			RADIOI UNLESS OTHERWISE NOTED			PARALLEL	<u>2-REQ'D</u>
		PERPENDICULAR		<u>2-REQ'D</u>								

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MARK PART NUMBER (X)		USED ON ASSEMBLY NO.		PROJECT NUMBER		OWENS-ILLINOIS, INC.	
TOLERANCE UNLESS OTHERWISE NOTED		DEFINITION OF SYMBOLS (Total Indicator Reading)		C-2331		CNPD - DEVELOPMENT CENTER	
2 PLACE DEC. ±		CONCENTRICITY		DES'D 11		TOLEDO, OHIO	
3 PLACE DEC. ±		ROUNDNESS		DR'N 11716 ED		MANIFOLD ASSEMBLY	
ANGLES ±		FLATNESS		CK'D 11		NUMBER	
CORNERS AND/OR EDGES BROKEN		SYMMETRY		1-REQ'D		SK-2997	
OUTSIDE RAD. MAX.		PARALLEL		SCALE FULL		SHEET 1 OF 1	
INSIDE RAD. MAX.		PERPENDICULAR		8		SIZE D	
UNLESS SPECIFIED, MACHINED SURFACE ROUGHNESS NOT TO EXCEED		RADII UNLESS OTHERWISE NOTED		9			
✓

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MARK PART NUMBER (X)			USED ON ASSEMBLY NO. SK-2331			CHG.	DESCRIPTION	BY	DATE	MF	
TOLERANCE UNLESS OTHERWISE NOTED	2 PLACE DEC. ± 0.010		DEFINITION OF SYMBOLS (Total Indicator Reading)	MATERIAL SEE ABOVE		PROJECT NUMBER C-2331		OWENS-ILLINOIS, INC. CNPD - DEVELOPMENT CENTER TOLEDO, OHIO MANIFOLD LINES			
	3 PLACE DEC. ± 0.005										
	ANGLES ±										
CORNERS AND/OR EDGES BROKEN	OUTSIDE RAD. MAX.		☒ CONCENTRICITY ☐ ROUNDNESS ☐ FLATNESS ☒ SYMMETRY	HEAT TREAT		DES'D	1 1				
	INSIDE RAD. MAX.			CASE DEPTH	DR'N	7/16 EL					
UNLESS SPECIFIED, MACHINED SURFACE ROUGHNESS NOT TO EXCEED	✓	RADII UNLESS OTHERWISE NOTED	☐ PARALLEL	HARDNESS		CK'D	1 1				
			☐ PERPENDICULAR	SUR. TREAT							
				2 REQ'D		SCALE	1/8 SIZE	SHEET	1	OF	1
										D	SIZE