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TOWARD ACCURATE PREDICTION OF COMPARATIVE FENESTRATION PERFORMANCE

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Toward Accurate Prediction of Comparative Fenestration Performance

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ABSTRACT

The importance of going beyond laboratory measurements to prediction of field performance of fenestration systems is noted. The current state of the art in energy prediction related to fenestrations is discussed; a critique of the ASHRAE U-value is presented, and current questions about film coefficients are put into context. Ambiguities in the modeling of complex glazing systems are pointed out. Several questions about the convective processes in simple enclosures are raised, and the importance and uncertainty of solar gain calculations is underlined. It is concluded that accurate field measurements are needed to inform the predictive enterprise. A facility to make these measurements is described and current progress on calibrating this facility is presented.

Introduction

This talk concerns future directions in fenestration research. I will therefore take as my starting assumption the successful achievement of the goal of this workshop, namely, an accurate, reproducible, and uniformly accepted methodology for making laboratory measurements of U-values for systems such as single or multiple glazing, sealed insulating glass, systems containing venetian blinds between glazings, etc. I term these types of systems <u>parallel-glazing</u> systems, because their interface with the environment on both the interior and the exterior is through parallel sheets of glass. Other types of systems, such as fenestrations with interior or exterior shades, blinds or shutters, I term <u>boundary</u> systems because they share the characteristic that the boundary with the interior and/or exterior environment is geometrically or thermally complex, and the concept of a film coefficient is not useful. It is instructive to note that most residential windows at night are in fact boundary systems, due to the occupants' desire for privacy.

A standard measurement method for the U-values of parallel-glazing systems is useful because it allows comparison between products; one wishes to do this comparison because U-value is related to energy performance. The next step is to elucidate this relationship and extend the comparison to include boundary systems and daytime performance. When this has been done it should be possible to predict the achievable energy costs or benefits resulting from a choice between alternative fenestration systems. This would allow the user of fenestration systems to make rational choices about energy performance, and to weigh this performance against other desired properties.

Accomplishing this goal will be an iterative procedure including both calculation and measurement. Standardized measurements on parallel-glazing systems can be used as input to existing techniques for calculating energy performance, such as the ASHRAE[1] procedures or the programs DOE-2[2] or BLAST[3]. These calculations should be compared with accurate field performance measurements. In addition, field performance measurements should be made on boundary systems and skylights. From these measurements new information will be obtained, first, on whether the calculation procedures are correct for parallel-glazing systems, and, second, on what modifications are necessary to include boundary systems, skylights, and the effects of solar gain. This information can be used to improve the calculation methods. Next, measurements and calculations covering a variety of conditions can be used to arrive at a set of representative conditions to be simulated in the laboratory for a uniform measurement treating all types of fenestration systems. The results of these laboratory measurements would then serve as input to the calculation procedures to provide the information needed by fenestration users.

Prediction of Fenestration Performance

The most widely-used simplified calculation method, and indeed the basis for this workshop, is the ASHRAE U-value, which is defined as the heat transfer per unit area per unit of difference between the interior and exterior air temperatures. Use of this quantity requires two assumptions: first, that the fenestration has a negligible heat capacity and, second, that all of the heat transfer processes are determined by the two air temperatures. The first assumption is non-controversial. The second assumption is a key one, since for parallel-glazing systems the interior and exterior glazing surface temperatures completely determine the heat transfer through the device itself, and hence its apparent conductance. If both of the parallel heat transfer processes of radiation and convection are assumed also to originate from a single temperature, then they may be combined into an effective "film coefficient" for the interior and the exterior surfaces. These are then combined with the conductance to obtain the U-value.

This U-value is, of course, not a constant, since it depends weakly on the temperature and more strongly on the air velocities at the interior and exterior surface. The ASHRAE assumptions for specifying these are of particular interest for the exterior surface. A value of 6.0 BTU/(hr ft^2 F) is typically taken as the "standard ASHRAE" condition; the origin of this value is somewhat uncertain. In Rowley, et. al.,[4] measured values of the forced convection coefficient for a wind-tunnel-produced wind blowing over a 1 ft^2 surface are given. If one uses their value for a glass surface and assumes interior and exterior air temperatures of 72 F and 32 F, respectively, one arrives at an overall outdoor film coefficient of 34.2 W/(m^2 K), or 6.0 BTU/(hr ft^2 F). This is a plausible explanation of the origin of that value. More modern work on forced convective coefficients[5] would imply a somewhat lower value of 31.5 W/(m^2 K) for the exterior film coefficient (under the same assumptions). The effect of this difference on overall U-value is negligible.

A frequently-raised criticism of the ASHRAE U-value is that the assumed 15 MPH wind speed is an extreme case and does not represent the average conditions seen by the window. Not only is the mean seasonal wind speed lower for most locations, the measured climatic data should also be corrected for height, surroundings, and shielding to obtain an estimate of the local wind speed.[6] For example, if these corrections are carried out for Baltimore, the winter-average wind speed of 4.5 m/s becomes a local average of 2.6 m/s for a one-story building. Measurements made by McCabe[7], not too far from Baltimore, found an average wind speed at the window of the NBS Passive Solar Test Facility to be about 1 m/s. This would imply a U-value for single glazing about 30% smaller than the ASHRAE value.

The foregoing considerations indicate that, even without questioning the basic physical assumptions of the U-value model, substantial uncertainties in energy prediction arise simply from the choice of "standard" conditions. To put this into perspective, if someone at the NBS location had invested in double glazing as an energy conservation measure, he would realize only 40% of the expected savings if these were predicted using the ASHRAE U-value.

There is, of course, no necessity to use such a simplified approach. Computer calculations such as DOE-2 and BLAST are much more sophisticated, treating interior air and radiative temperatures separately, determining them through a net heat balance, and adjusting outdoor film coefficients to account for wind and (possibly) sky and ground radiation. It has sometimes been argued that, given these sophisticated theoretical treatments and the fact that the individual (convective and radiative) heat transfer coefficients are known to be slowly-varying functions of temperature and air flow rate, even quite a simplified model of the internal and external conditions will yield accurate predictions of heat flow.

Two objections to this point of view must be raised. First, even if this argument were true for parallel-glazing systems, substantial uncertainties remain for boundary systems. For example, for an interior shade, shutter, or blind, two different models are possible depending on whether one considers the mixing of air between the device and the glazing with the interior air to be significant. A similar problem arises in considering an exterior shutter or closed blind in the presence of wind: What should be assumed for the air between the shutter and the glazing? Should the heat transfer between the air and the glazing be considered turbulent, with the air at the external air temperature? Should one assume laminar-forced convection? Natural convection? Each set of assumptions will lead to a significantly different prediction of the overall heat transfer.

A second objection, suggested by recent research on natural convection in enclosures, is both more subtle and more far-reaching. The use of convective film coefficients for building interiors models the heat transfer as flowing between surfaces and a well-mixed, turbulent core through a thin boundary layer. It is clearly a picture founded in forced convection. The concept of convective heat transfer coefficient is, of course, a rigorous one with firm basis both theoretically and experimentally for either forced or natural convection: It determines the amount of heat flow between a fluid (in this case, air) and an adjacent solid. However, nothing in the concept determines the ultimate source (or destination) of the heat in the fluid; that is determined by the overall character of the fluid flow outside the thin conductive boundary layer. For example, in natural convection at a heated vertical surface the heat flows through the thin conductive boundary layer into an upwardly-moving boundary-layer flow which is relatively isolated from the adjacent still-air core. If the heated surface formed one wall of a building space, the initial heat transfer would be upward; whether and to what extent the heat were ultimately transferred to the bulk of the room air would depend on the extent to which this boundary flow of air eventually mixed with the air in the core.

This observation gathers force in the light of research which suggests that laminar natural convection may be important in building spaces. Both experiments[8,9] and numerical simulations[10] have indicated that even for the large Rayleigh numbers characteristic of building spaces, convection remains laminar when the important hot and cold surfaces are on walls, while scale model studies[11] have indicated that even for the case of local heating of the floor (as from a patch of sunlight) the induced convection pattern is very far from complete turbulence.

This raises interesting possibilities for the thermal behavior of rooms with windows. Consider, for example, a room containing a window and a radiator. The heat from the radiator (which is really a convector) will be carried upward by a (probably turbulent) air plume, which will cause a stratified layer of warm air at the ceiling. The thickness of this warm layer would grow until its lowermost edge impinged on the window, at which point a convective flow of cool air would be set up. Cold air from the window would in turn cause a layer of colder air at the floor, which would grow thicker until it in turn reached the radiator. Clearly an equilibrium is possible, in which there is a recirculating convective flow carrying heat between the radiator and the window. The core of the room (containing presumably the thermostat) would be relatively isolated from this convective loop if the flow remained predominantly laminar. In this picture, although the film coefficients at the window would be approximately what one would expect from theory, the effective temperature driving the heat flow would be that of the circulating loop, which could differ significantly from the core temperature. A very approximate model of this type, assuming reasonable temperatures for the radiator and the convective loop, indicates that the actual heat transfer through a single glazed window might be 20% larger than expected from the usual calculation method.

This example points to what may be a general class of heat transfer processes in rooms not treated by the usual theory. To the extent that laminar natural convection exists, it will seek out and connect the hottest and coldest surfaces in the room. The heat flowing through this recirculating loop will be only weakly dependent on the bulk air temperature of the space. This includes the potential for carrying solar heat deposited in walls or floors back out through the window which admitted it, rather than having it eventually transferred to the interior air as generally is assumed. Moreover, these processes could even connect windows with heat sources in different rooms, providing there were connecting openings.

Another possibility is the following model for a room in a residence utilizing forced-air heating. Consider the instant in the heating cycle at which the furnace switches off. Because the room air has been thoroughly mixed by the heating system, one can consider the air to be uniformly at the upper temperature of the thermostat dead band. The air will come to rest, and heat loss through the window and walls will set up laminar convective flows. The cold air will accumulate on the floor, since there is no source of heat in the room. The layer of cold air then builds up until its upper edge reaches the thermostat and causes the furnace to turn on.

Whether these models, which are clearly speculative, reflect situations which actually occur in practice is impossible to say at present. One would need to know the time constants for convective processes to be set up, the degree of mixing and air movement in actual building spaces such as residences, etc., none of which information exists. One would guess that such processes would be of less importance in commercial buildings, provided their HVAC systems work as designed. It is clear, however, that reality may be somewhat different from the picture assumed in our calculation models, and that there is no guarantee that the model even represents a correct first approximation for all building situations.

Hitherto our discussion has centered around U-value, which is to say, nighttime performance. In a 1975 study,[12] however, it was pointed out that the seasonal energy costs or benefits to be associated with windows are substantially affected by the solar gain that is (intentionally or inadvertently) collected by the window. A more detailed study[13] conducted by our group indicates that there is an optimal combination of U-value and shading coefficient, from the standpoint of annual energy use, and this optimum varies with location and orientation of the window. While these studies point to great opportunities for improved energy performance through the optimal use of fenestration, they are so far based entirely on calculation and need experimental verification. Key questions are how much of the solar gain is retained and is beneficial, and what modifications to the heat transfer processes occur in the presence of sunlight.

Measurement of Fenestration Performance

The foregoing discussion points to the importance of accurate and reliable measurement of fenestration performance under realistic conditions. These measurements are necessary to check the predictive techniques used to calculate performance, to resolve ambiguities in the correct approximate models, and to provide input data to the models once they have been perfected.

While measurements of windows have been made on numerous occasions, both in buildings and in passive test cells, the uncertainties inherent in these measurements make it difficult to combine them into a definitive body of information.[14,15] This is especially true for systems with substantially higher thermal resistance than single glazing. To create a base of accurate data a rather specialized facility is necessary. The generic characteristics desirable in such a facility are listed in Figure 1.

A facility of this type, the Mobile Window Thermal Test (MoWiTT) Facility, has been built at Lawrence Berkeley Laboratory and is currently being calibrated. Because detailed descriptions of the facility have been published elsewhere,[16,17] I shall give only a brief description and concentrate on our current state of progress.

The MoWiTT measures the net energy flow through a fenestration system, including beam or diffuse solar gain when present, by determining the net heat balance on the adjacent room-sized space, as shown in Figure 2. The fenestration heat flow, W, is then calculated from the equation

$$W(t) = C\,V \frac{dT}{dt} - H(t) - I(t) + L_C(t) \qquad (1)$$

All of the quantities on the right-hand side of this equation are measured.

A critical issue is the experimental uncertainty in the measurement, which is given by

$$\frac{\delta W}{W} = \left\{ \left[\frac{\delta(C\,V \frac{dT}{dt})}{W} \right]^2 + \left[\frac{H}{W}\left(\frac{\delta H}{H}\right) \right]^2 + \left[\frac{I}{W}\left(\frac{\delta I}{I}\right) \right]^2 + \left[\frac{L_C}{W}\left(\frac{\delta L_C}{L_C}\right) \right]^2 \right\}^{1/2} \qquad (2)$$

The manner in which this issue has determined the design of the MoWiTT, together with the expected performance, is given in reference 14.

The overall design of the MoWiTT is shown in Figure 3. It consists of one or more self-contained measurement modules, together with a central instrumentation station where data is collected and monitored. Each measurement module contains a pair of identical room-sized calorimeters. A removable sample-holding wall on each calorimeter permits mounting a fenestration system up to approximately 2.3 m by 2.3 m in size. The paired calorimeter arrangement permits accurate comparison measurements to be made using actual weather conditions. The mobility of the module allows one to select different orientations and climates.

The MoWiTT has two distinguishing features that derive from the accuracy requirements for measuring advanced fenestrations and set it apart from other field measurement facilities. These are an air-guarded calorimeter in combination with large-area heat flow sensors.

The function of the air guard around the calorimeters is to isolate them from the exterior environment, which would otherwise introduce large uncertainties through the time-varying, thermally-driven envelope heat transfer. The guard system, shown in Figure 4, circulates controlled-temperature air around all sides of the two calorimeters except for the region containing the fenestration samples, which bridge the guard space. (Fenestration samples may be either windows or skylights.)

The function of the heat flow sensors is to measure the inhomogeneous, time-dependent envelope heat flows due to heating of the interior envelope surfaces by admitted solar gain. Because of the spatial inhomogeneity and time dependence of the solar flux, essentially complete coverage of the interior calorimeter surfaces is necessary. This has necessitated the development of economical and accurate large-area heat flow sensors, shown schematically in Figure 5. These have been described in detail elsewhere.[18,19] The sensors are produced in various sizes, normally about 0.5 m X 1.0 m, and are connected in series to form effectively wall-sized sensors. A calorimeter chamber instrumented with them is shown in Figure 6.

The first measurement module of the MoWiTT, its staff, and a snapshot of activities in its control room are shown in Figures 7, 8 and 9, respectively.

The first calorimeter chamber of the facility is currently being calibrated. The purpose of the calibration is to demonstrate empirically that the net heat balance measured by the facility is correct, and to document experimental biases and sensitivities, so that reliable accuracy estimates can be made for measurements on fenestrations. The MoWiTT will be the first fenestration measurement facility for which such information will be available. It must be stressed that if measurements are to be used in any systematic way, such as to validate computer algorithms, to compare performance at different locations, or to develop methods of computing average energy costs, this type of accuracy information is vital.

The method used in the calibration is to replace the window-holding sample wall with a double calibration wall which allows complete flow of guard air around the calorimeter chamber. The inner calibration wall is covered with heat flow sensors in the same manner that the sample wall will be covered during measurement, except that the area which would be occupied by the fenestration sample is also covered with (separately recorded) heat flow sensors. This forms the calorimeter chamber essentially into a closed box, all sides of which (except the common wall with the adjacent calorimeter) are surrounded by guard air. All heat added or removed by the temperature control system is measured in the normal manner, and any heat flowing through the envelope is measured by the heat flow sensors. All of the heat flows measured for the chamber should therefore sum to zero. The guard and the neighboring calorimeter are kept at the same temperature.

Since inhomogeneous solar fluxes are an important condition to be measured by the MoWiTT, it is important to establish that the location of a radiant flux does not affect the heat flow sensor measurement. To check this, spotlights were inserted into the chamber and directed at various locations on the walls where one might expect different sensor response, i.e., the center of a sensor, the edges and corners of walls. By switching these lights on and off sequentially, a moving radiant flux can be simulated.

The results of a preliminary set of these tests are shown in Figure 10, 11 and 12. In Figure 10 the temperatures of the calorimeter being calibrated (Chamber A), the adjacent calorimeter (Chamber B) and the guard are shown during the period of the test. As can be seen, the temperatures remained quite constant, allowing one to neglect heat capacity terms in the energy balance equation. The small excursions in the Chamber A temperature mark the transient effects of changing conditions between successive tests. During this test the air temperature of the guard and chamber B were kept low to maximize the heat flow through the envelope, in order to test the heat flow sensors. This does not represent the normal operating conditions for the calorimeter.

Figure 11 shows the individual heat flows into and out of the calorimeter. (Negative heat flow denotes heat flow outward.) The test begins just before day 2, at which point the input heater is stabilized. After a sufficient time has passed to allow all the chamber subsystems to come to thermal equilibrium (since radiative temperatures may change even though air temperatures do not). the first lamp is turned on and the input heater power decreased to compensate for the additional power input. Subsequently, power is switched sequentially between lamps at different locations.

Except for transient excursions, some of which may be due to electronic noise associated with switching, as well as to thermal transient effects, the total heat flow measured by the heat flow sensors remains quite constant, independent of the source or location of the heat input to the chamber. The small change in the heat flow sensor signal visible in Figure 11 is due to the small shifts in guard and chamber temperatures visible in Figure 10, rather than to changes in the source location.

In Figure 12, these heat flows are added up as they would be during a fenestration measurement. This yields a measurement, derived from the net heat balance on the chamber, of the energy flowing through the area of the calibration wall where a fenestration sample would normally be placed--in effect, that section of the calibration wall is treated as a window with very high R value. However, since that area is also covered with heat flow sensors we have a direct measurement of this heat flow as well. The two measurements are plotted in Figure 12.

As can be seen, the two measurements differ by some 30 W, which may be taken as the level of accuracy of the facility at this stage of calibration. This is quite an acceptable level of accuracy for many glazings and conditions of interest; however, the source of this difference will be determined before high R-value measurements are attempted. It appears currently that this accuracy should be much better under normal running conditions, when the guard and chambers are all kept at the same temperature.

We consider these test results quite encouraging. They are not, however, conclusive because for technical reasons the thermal fluxes used had to be kept much smaller than would be experienced under solar gain conditions. We plan to repeat the tests with realistic flux levels in the near future.

Conclusions

Computer simulation of fenestration performance under realistic conditions, while it holds promise, cannot at present be said to provide all the answers. It needs verification and resolution of model ambiguities which can only be provided by examining accurate data measured under representative conditions. A facility to make these measurements, the MoWiTT, is presently completing calibration and should begin providing this data in the near future.

Acknowledgement

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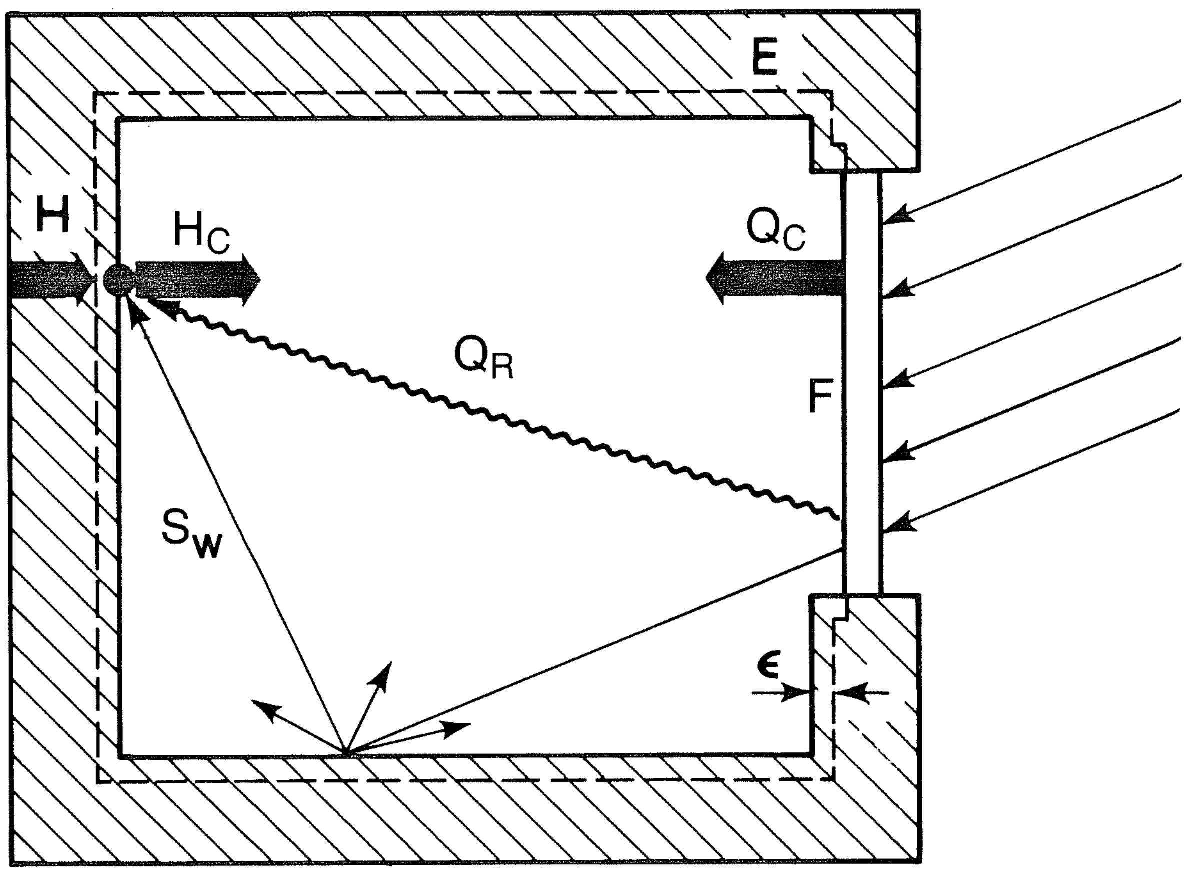
REQUIREMENTS FOR IDEAL DEVICE TO MEASURE WINDOW AVERAGE PERFORMANCE

- Real weather
- Wide range of climatic conditions
- Good recording of exterior conditions
- Good control for weather variability
- Variable orientation
- Well controlled interior environment
- Full-size, room-like interior space
- Building of variable characteristics
- Calorimetric accuracy in measuring heat flows
- Highly instrumented

Figure 1. Desirable measurement facility characteristics.



XBL 847-10694

Figure 2. The MoWiTT measures the net energy flow through a fenestration system, including beam or diffuse solar gain when present, by determining the net heat balance on the adjacent room-sized space.

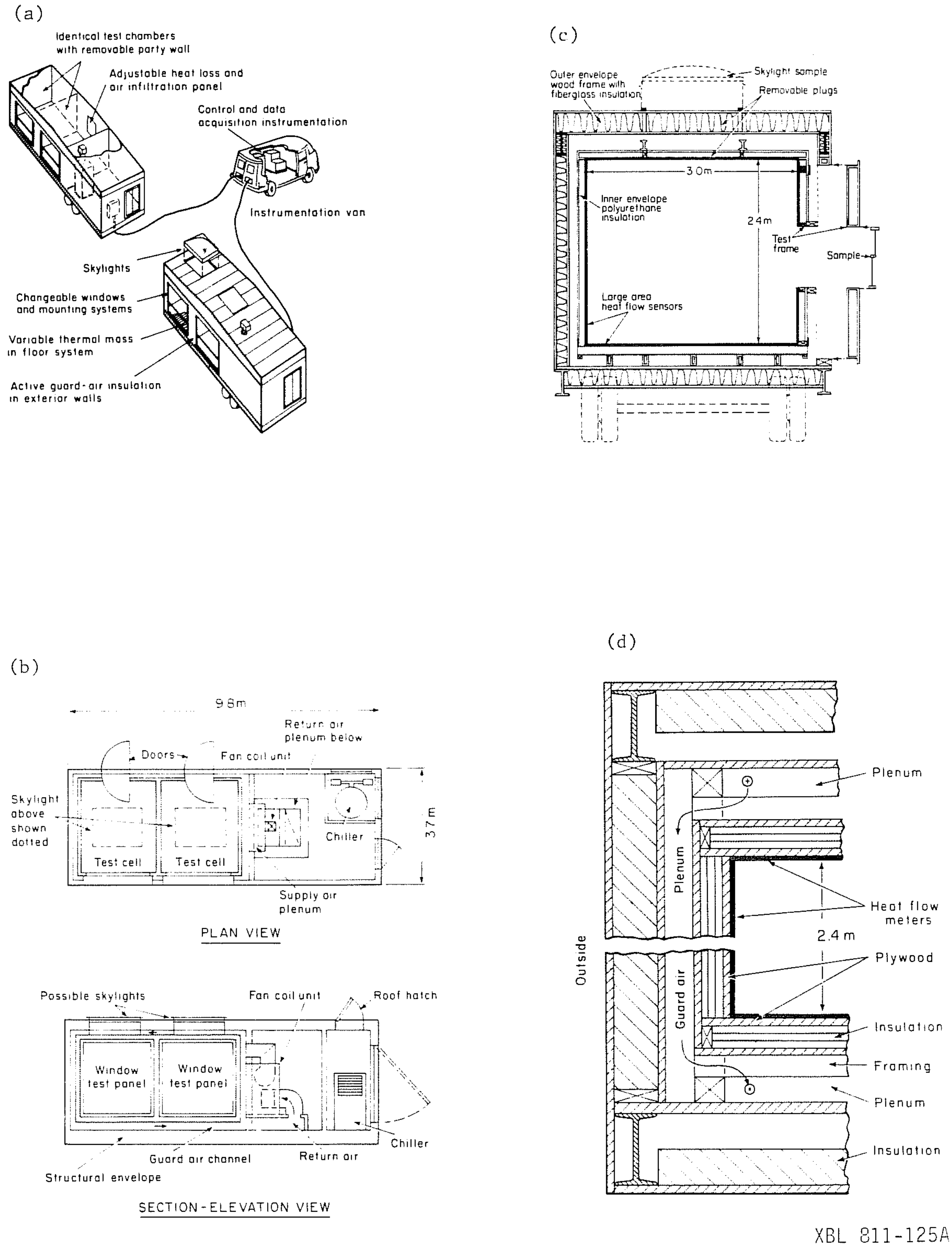


Figure 3. Overall design of the MoWiTT.

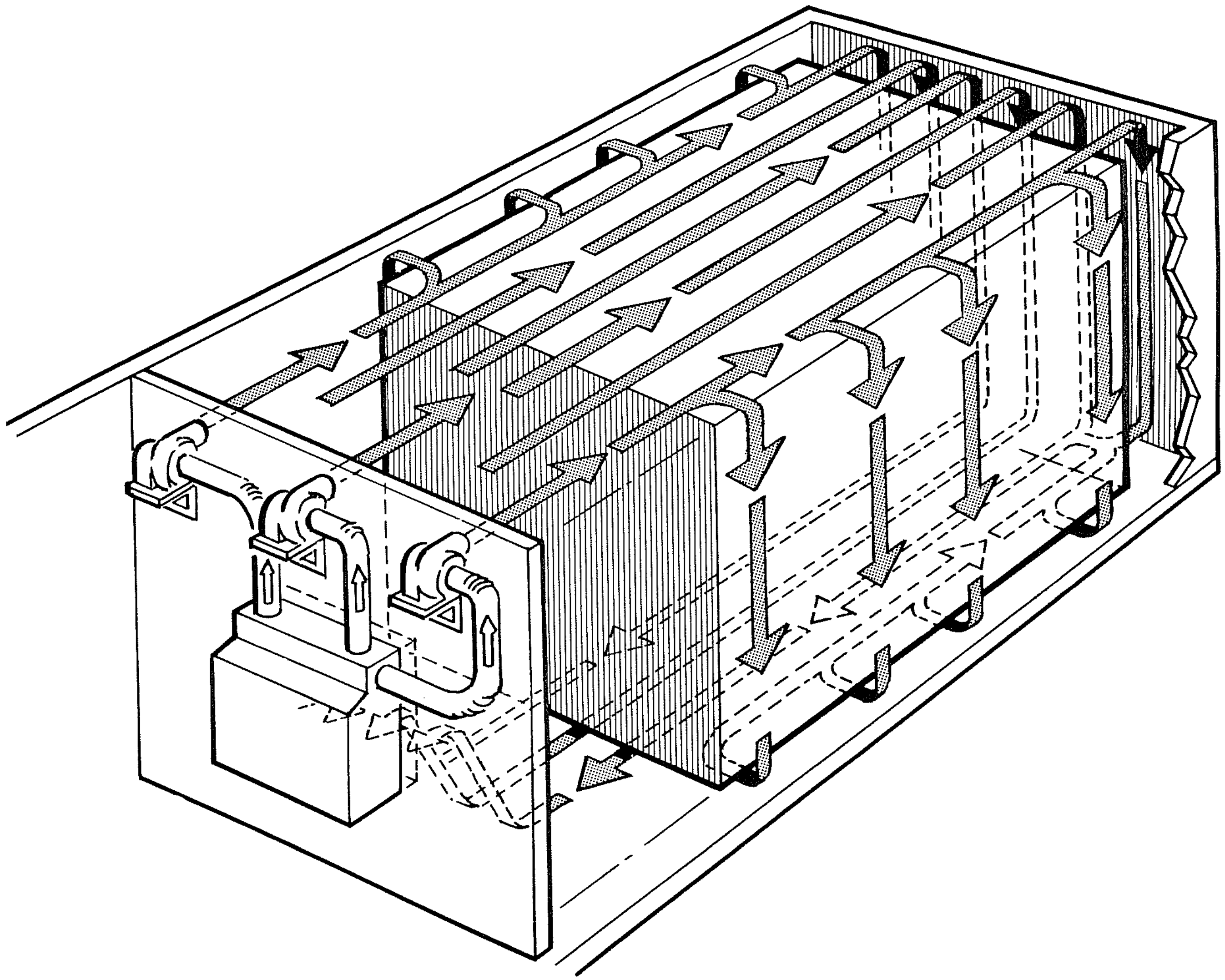


Figure 4. MoWiTT guard air flow system.

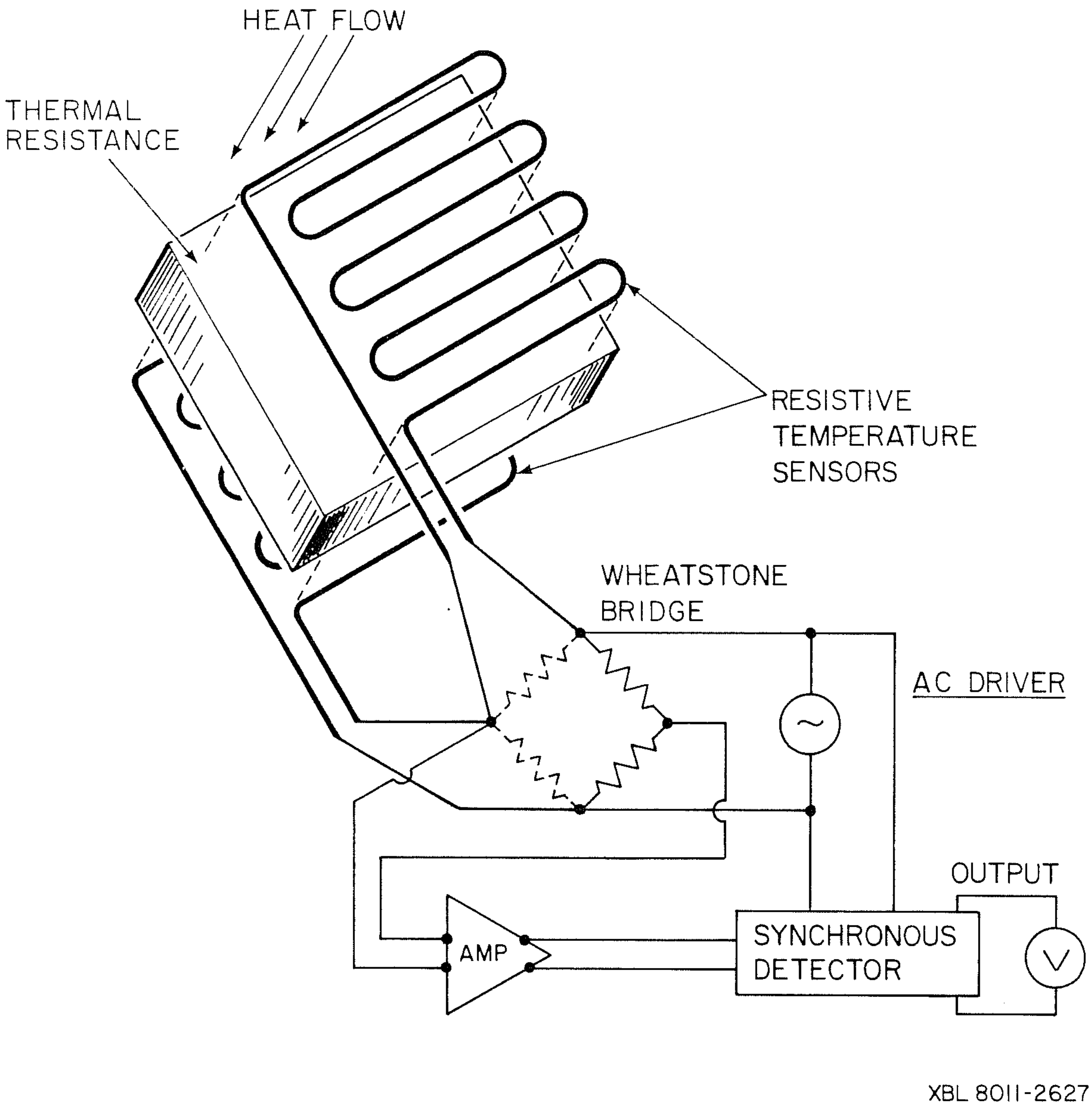


Figure 5. Large-area heat-flow sensor.

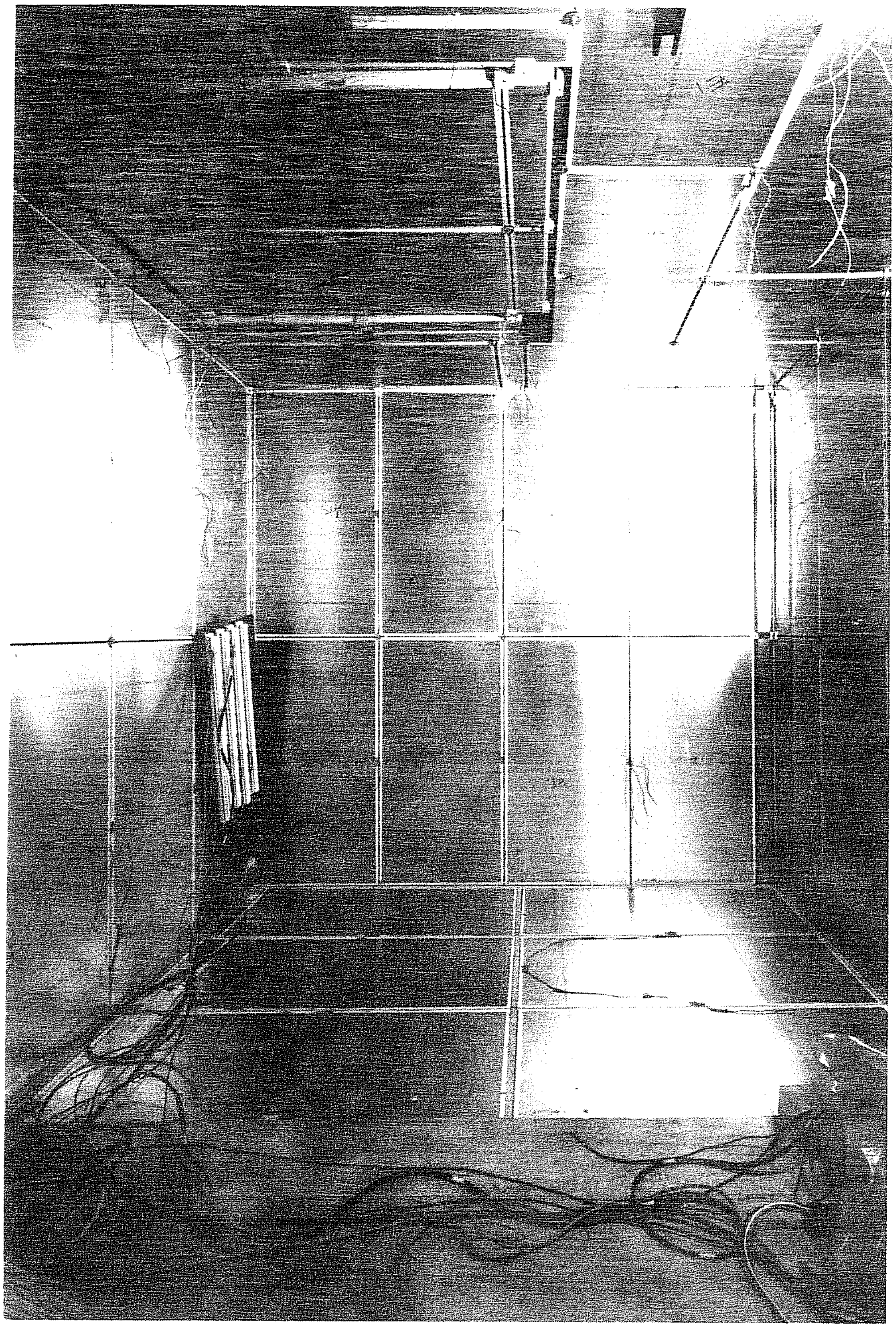


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Figure 6. Interior of calorimeter chamber instrumented with large-area heat flow sensors. More realistic wall and floor treatments are added prior to measurement.

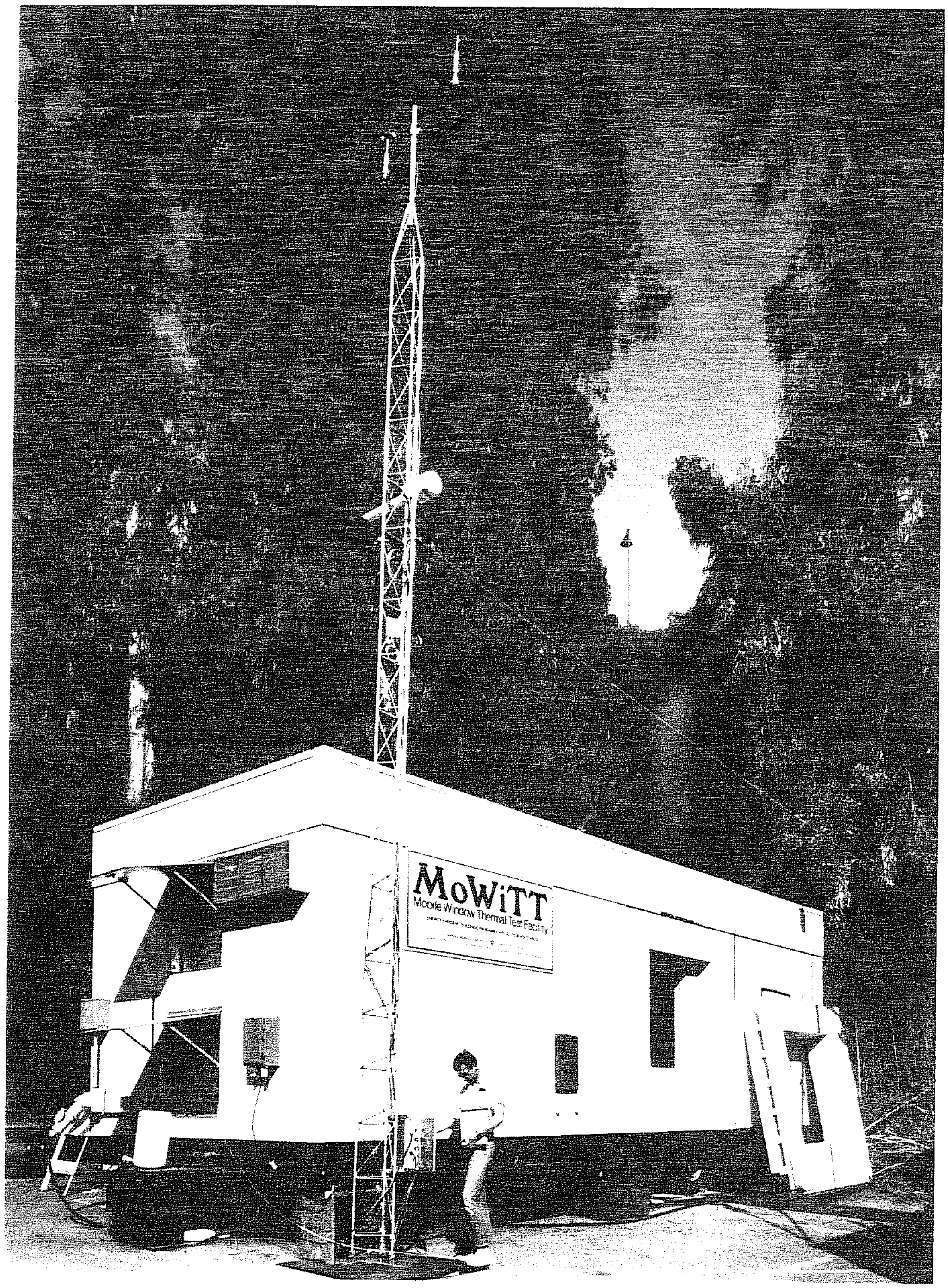

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Figure 7. The first MoWiTT module during calibration at Lawrence Berkeley Laboratory.

XBB 847-6260

Figure 8. The MoWiTT experimental staff.

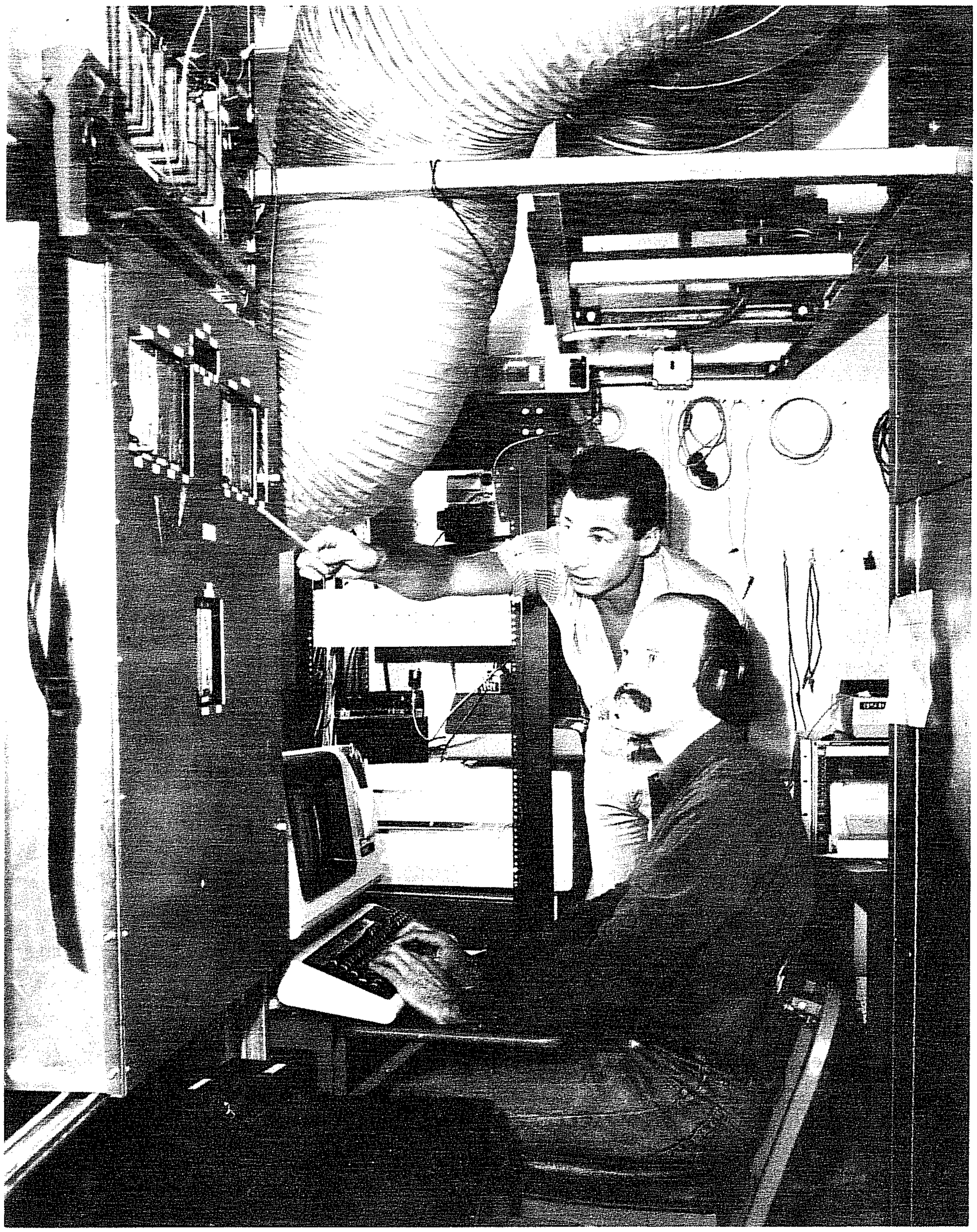

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Figure 9. Two members of the MoWiTT staff discuss settings for the calorimeter temperature control system. The CRT unit is used to control and monitor data collection by a real-time computer system.

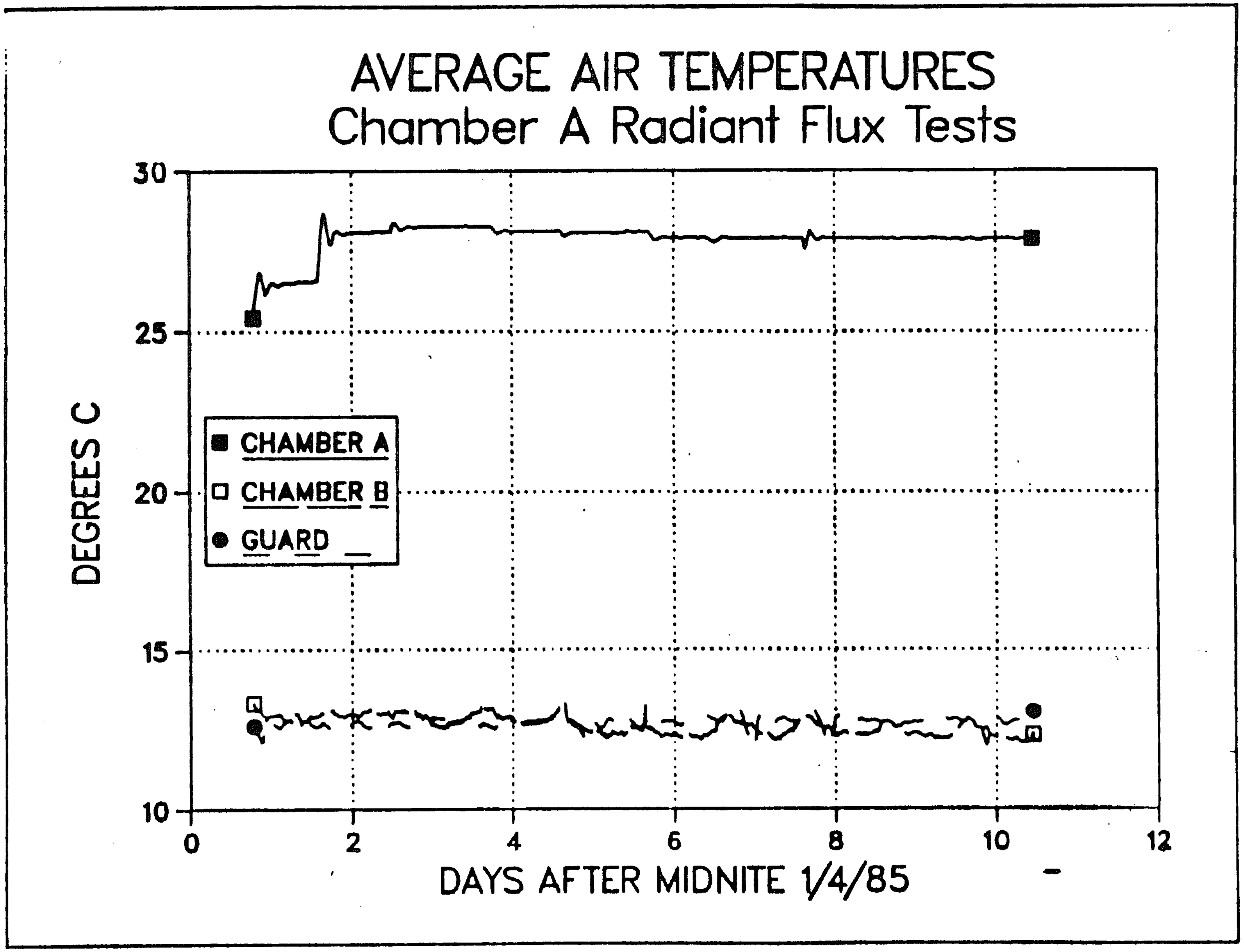

Figure 10. Temperatures during calibration of Chamber A.

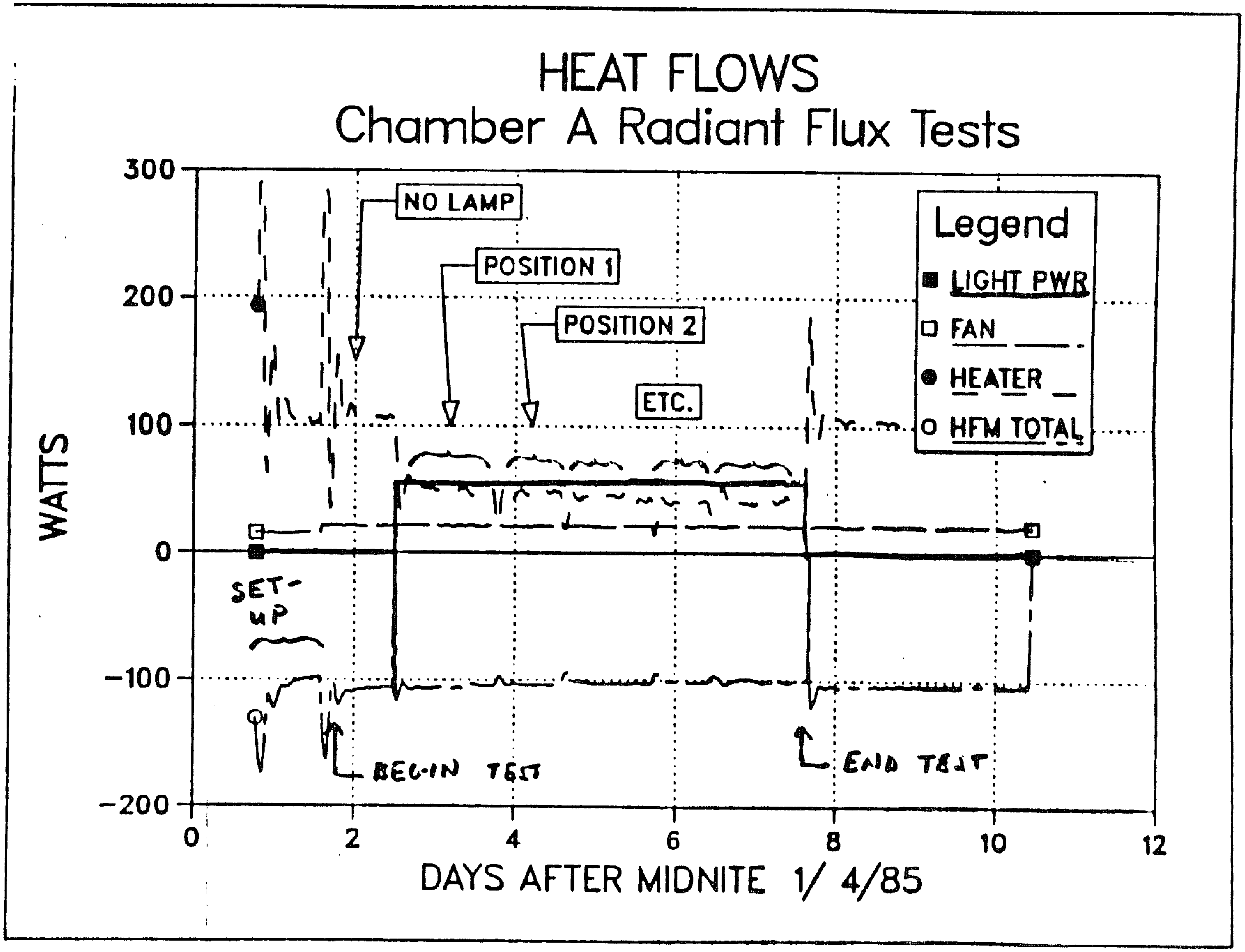

Figure 11. Heat flows during calibration of Chamber A.

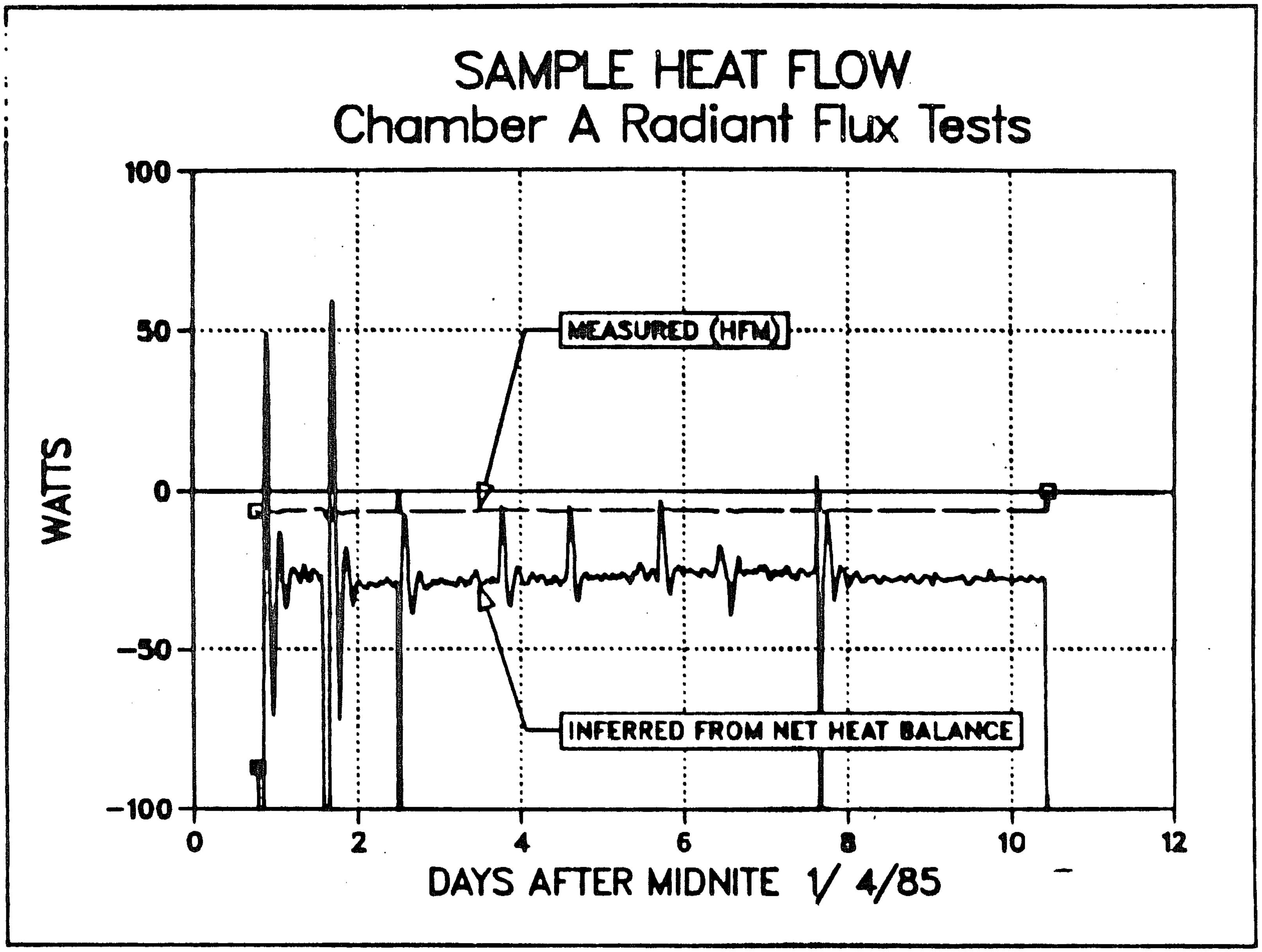

Figure 12. Measured heat flow through calibration sample compared with heat flow inferred from the measured net heat balance on Chamber A.