AN EXPERIMENTAL STUDY OF COOLANT COMBUSTION EFFECTS IN TRANSPIRATION COOLING

by

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A THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1970

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PART I

INTRODUCTION

Two possible causes of high convective heating are viscous dissipation in the boundary layer and high energy fluid streams. A space vehicle reentering the atmosphere of the earth is the most widely known example for the first cause, and turbine vanes and blades or rocket nozzles are exposed to the second.

One way to protect structures from high convective heating is by application of mass transfer cooling. The basic principle of mass transfer cooling is the displacement of material from a surface which can be achieved in three fundamentally different ways; 1) by ablation, 2) through film cooling, or 3) transpiration cooling.

The thermal erosion of material at high temperatures is known as ablation. The cooling with ablation is accomplished through the heat sink properties of the ablating surface, thus shielding the structure underneath from high heat flux. Additionally, the ablation product emerges into the boundary layer, where it in effect forms an insulating fluid film with the result of a reduced heat convection to the structure.

Reduced heat convection can also be accomplished by injecting a coolant material through discrete slots or holes to form a fairly uniform, continuous, insulating film over the outer surface of the structure in need of protection. This mass transfer cooling is called film cooling.

In transpiration cooling, the coolant material is forced through a porous wall to provide cooling for the surface exposed to high convective heating. The thermal protection is achieved in two ways:

1. The coolant, in passing outward from the reservoir through the porous wall to the surface to be cooled, absorbs heat from the porous wall. This convective cooling is extremely effective due to the high surface to volume ratio of the porous material, even with a relatively small coolant flow.

2. The coolant, after reaching the surface of the porous wall, emerges relatively well distributed into the external boundary layer and thus alters this boundary layer in such a manner as to decrease the heat transfer to the wall. This is refered to as "blockage" effect in some literature.

Several studies have revealed the superior effectiveness of transpiration cooling compared to any other cooling method [1, 2, 3, 4]*. The most common application for transpiration cooling so far has been the cooling of gas turbine vanes and blades [5]. The feasability of a transpiration cooling system for reentry vehicles, which are until this date the domain for ablation, has been demonstrated in a study by Jaeck and co-authors [6] in 1968. Terry and Caras [1] investigated the transpiration cooling of liquid rocket nozzles.

In transpiration cooling, the porous wall is exposed to a mainstream flow. The coolant fluid flowing through the porous wall is the secondary stream. Transpiration cooling has been accomplished in the following combinations of main and secondary stream:

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*Numbers in brackets designate references in SELECTED REFERENCES.

1. Coolant and mainstream fluid have identical properties,
2. Coolant fluid is a foreign gas,
3. Coolant fluid is a liquid.

The coolant fluid may be an inert material, and others are capable of undergoing a chemical reaction. Some coolants will decompose endothermically (e. g. endothermic dissociation) at a certain temperature range. The products of dissociation have mostly lower effective molecular weights, and generally a higher specific heat than the reactants. The increased heat absorption capacity and a self-regulating feature for the endothermically reacting coolant fluid in transpiration cooling have been demonstrated by Rosner [3] and Gorton [7].

In a survey of publications concerning transpiration cooling prior to 1960 Leadon [2] shows that low molecular gases appear to be superior transpiration coolants. This has been attributed to the more effective heat blockage of lighter gases. However, gases with low molecular weights tend to have relatively high specific heats, thus permitting a greater heat absorption through the porous wall, which may very well account for a part of their increased effectiveness.

For gas turbine cooling purposes air has traditionally been selected as the coolant fluid. This was done since air is readily available, and it requires only a minimum of system complication. With increased power requirements for gas turbines their inlet temperature has been increased concurrently, resulting in a demand of larger quantities of cooling air to maintain a safe temperature level for structural parts. The experimental study in reference [5] applied approximately up to 11% of the compressor

flow for cooling. One recent proposal calls for up to 25% of compressor inlet air for cooling purposes [8]. For some applications the latter is excessive and alternate cooling fluids should be considered. The most obvious alternate coolant is the fuel. Using the fuel for cooling eliminates the provision of a special coolant supply. Some applications where the cooling with fuel is particularly attractive are the engines of the proposed space-shuttle reentry vehicle, the gas turbines driving the fuel pumps for hydrogen-oxygen rockets, and liquid methane fueled aircraft engines.

Transpiration cooled turbine vanes and blades are especially suited to use of fuel-coolants. The fuel is not wasted after its use in cooling, because its energy is available to the mainstream flow which it enters. However, combustion of the fuel-coolant could adversely affect the effectiveness of the transpiration cooling system. It is conceivable that boundary layer combustion could actually result in increased, rather than decreased, temperatures for "cooled" surfaces.

The purpose of this study was to provide preliminary information on the influence of boundary layer combustion on transpiration cooling effectiveness. The study was limited to a single surface under one set of mainstream flow conditions. The coolants used were hydrogen, ammonia, and nitrogen. Hydrogen was chosen because of its presence as a fuel in some of the proposed applications and because of its desirable properties (i. e. low molecular weight, high specific heat) for transpiration cooling. Ammonia was chosen because it is a clean and carbonless coolant and because of its potential for endothermic dissociation at high temperature [7]. Nitrogen was selected as a non-reactive comparison coolant.

PART II

EXPERIMENTAL SYSTEM

1. Experimental Apparatus. The arrangement of the experimental system used is shown in Figure 1. The hot gas source for the mainstream was a "Wondaire" gas burner, model 260B, fueled with propane gas. The heating stream flow was directed through a hot gas duct, 3.5 inches wide, 5 inches high, and 24 inches long. Upper wall and both side walls of the duct were made of 0.016 inch thick stainless steel. The porous surface formed a 3 by 6 inches portion of the lower duct wall, with centerlines coinciding. The forward portion of the lower duct wall, extending 11 inches from the duct entrance, was built from a steel plate, 0.187 inch thick, with an 1 inch layer of asbestos on the outside. The next section was the test surface with plenum chamber surrounded on all four sides by asbestos board, 0.25 inch thick. The final 6.5 inches length of the lower duct wall were made of 1 inch thick asbestos. The duct wall above the porous specimen was perforated at several spots along the centerline for the insertion of pitot tube and thermocouple.

To increase the mainstream velocity the hot gas duct was placed in a 24 by 24 inches exhaust duct with an exhaust fan at its other end. The cooling water from the cover plate was sprayed into the hot gas stream coming out of the hot gas duct to prevent the overheating of the exhaust fan.

The velocity of the mainstream was determined from measurements taken with a 0.125 inch diameter pitot tube and indicated by a micro-manometer. The locations for the measurements were 0.75 inch above the porous surface at the leading edge, 1.4 and 4.4 inches from the leading edge along the centerline. The temperature of the heating stream flow was obtained with an unshielded Chromel-Alumel thermocouple at the same locations as was the pitot tube. The voltage output from the thermocouple was measured with a Leeds & Northrup, model 8686, millivolt potentiometer with an ice bath reference junction.

Figure 2 shows the plenum chamber in a sectional sideview. The 3 by 6 inches porous test plate was constructed of seven layers of 12 x 64 mesh wire cloth (N-155 alloy, "Rigimesh") rolled to 0.25 inch thickness and welded to a 1 x 3 x 6 inches plenum chamber made of 0.016 inch N-155 material, with an one inch wide base flange on all four sides. A water cooled cover plate made of 0.5 inch thick steel, insulated by a 0.25 inch thick asbestos board, was bolted to the base flange of the plenum chamber. Four steel strips were used to obtain an even clamping of the thin base flange, and excellent sealing was achieved with 0.063 inch thick asbestos gaskets underneath the base flange and between asbestos board and cover plate. The bundle of thermocouple wires was potted into a short stainless steel tube with Sauereisen, No. 1 Paste and sealed in the cover plate with standard tube fittings. A 0.25 inch diameter copper tube perforated to distribute the coolant gas over the length of the plenum chamber was fitted through the cover plate and brazed for sealing. Openings for thermocouple leads and for the coolant inlet tube were provided also in the lower asbestos gasket and board. The bottom of the plenum chamber was covered by two polished stainless steel radiation shields.

Chromel-Alumel thermocouples were imbedded in 0.063 inch diameter holes approximately 0.03 inch deep in the lower surface of the porous material, secured by discharge welding and covered with Sauereisen, No. 1 Paste. These were placed at one-inch intervals along the plate centerline, beginning 0.5 inch from the leading edge. Additional thermocouples were placed one inch on either side of the 3.5 inch centerline location to check temperature gradients transverse to the flow direction. One thermocouple was installed approximately 0.03 inch below the upper plate surface with ceramic tube insulation to allow the determination of temperature difference across the plate thickness at the 1.5 inch centerline location.

A thermocouple was placed in the perforated copper tube for the temperature determination of the coolant gas brought into the plenum.

Coolant flow rate was determined from an overlapping range set of three 100 mm variable area flow tubes. These were calibrated with a positive displacement gas meter for hydrogen and nitrogen. Calibration for ammonia was determined with an orifice arrangement which had been previously calibrated for nitrogen by use of the gas meter.

Additional instrumentation included thermocouples on the hot gas duct above the porous testplate and on the upper sides of the two radiation shields. These thermocouple junctions were also covered with Sauereisen, No. 1 Paste. All thermocouple outputs from locations on the porous specimen were recorded on a 20 point recording potentiometer, Speedomax Type G from Leed & Northrup. An ice bath reference junction was used. All others were recorded manually from millivolt potentiometer indications.

2. Comments on the Apparatus and Instrumentation. The final apparatus design was the result of experience gained from several prior experiments. First attempts used a sintered powder porous material. This type material seems ideal for transpiration cooling for low temperature applications. However, experience has shown that it is totally unsuited for high temperature uses. It invariably failed from thermal stress; in some cases it disintegrated into a powder at local hot spots [9]. The wire mesh material used in the final design of this experimental investigation proved mechanically superior to the sintered materials. No structural difficulty was experienced with it in the system.

Earlier designs had thermocouples at both the upper and the lower surfaces. However, in experiments with a 0.125 inch thick wire mesh plate, it was found that the temperature difference across the plate thickness was a constant which depended on coolant flow rate, but which was practically independent of location. Because of this and due to the difficulties of installing the upper surface thermocouples without introducing coolant leaks, the final design included but a single upper surface thermocouple. This thermosouple was located approximately 0.03 inch below the upper surface rather than on the surface.

Because of operational difficulties it was decided to use a bare thermocouple junction for the temperature measurements of the heating mainstream. The radiation error introduced was estimated by calculation to be 7% or 150 degrees. For comparison, the maximum variation experienced among all measured mainstream temperatures at any single location was 150 degrees. Because of the difficulty in obtaining accurate temperature and velocity pressure measurements under the severe conditions of the experiment, nitrogen was used as a nonreactive comparison or reference coolant. All results

can be discussed in terms of nitrogen results under identical experimental conditions.

3. Experimental Procedure. For each set of experiments with all coolants, a low flow of nitrogen was first started to the coolant plenum chamber. The propane burner was fired, the test coolant flow was introduced at the desired flow rate, and steady state conditions were obtained. Steady state was determined from observation of the plate thermocouple indications on the recording potentiometer. Heating stream temperature and velocity pressure were measured at three locations along the plate centerline. The instrumentation was removed and all manually recorded temperature outputs were noted.

The coolant flow rate was then changed to a new value, steady state was determined and the above procedure was repeated.

A nitrogen purge of the coolant lines and plenum chamber was used before and after each series of hydrogen and ammonia tests.

For the zero blowing runs no coolant was flowing into the plenum chamber, steady state conditions were obtained and temperature and velocity pressure were measured. This was repeated before and after each set of experiments.

In operation the system was very stable. Test conditions and experimental data were reproducible over the several weeks of testing for all coolants and for the zero blowing runs.

PART III

RESULTS OF THE EXPERIMENTS

1. Experimental Results. The temperature of the mainstream during the test was 2100 °F. As noted before, the maximum temperature variation was 150 degrees. The velocity of the mainstream during the test was 110 fps with variations within $\pm$ 7% of that value. Variations among values at the three measured points along the centerline of the plate were insignificant.

The temperature difference between lower and upper surface of the porous test plate was assumed to be constant over the plate, as it was noted earlier. Measurements of temperatures at the upper surface (T_W) and the lower surface (T_L) at the centerline location 1.5 inches from the leading edge are plotted as a function of the dimensionless blowing rate ($F=(\rho V)$ coolant/(ρV) mainstream), Figure 3. It can be seen that the temperature difference depends slightly on the blowing rate and seems to increase with increasing specific heat of the coolant gas. This is in agreement with an analysis by Green [10], in which the temperature difference across the wall is shown to be proportional to the product of coolant flow rate and specific heat for constant porous wall properties.

The experiments were limited to blowing rates of $F<0.002$ for hydrogen and $F<0.006$ for ammonia because of limitation by the range of the flow meter set. For nitrogen the experiments could be carried out to $F=0.0074$ and the results obtained confirm the trends shown only to $F=0.006$.

Temperatures measured along the lower plate surface for nitrogen, hydrogen, and ammonia runs are shown in Figures 4, 5, and 6, respectively, for all blowing rates accomplished. On each plot the temperature distribution for the zero blowing is also shown. The heavier curves represent temperatures from approximately equal increments of the blowing rate; lighter curves are results of closely spaced low blowing rate tests made to enable a better comparison of the effect of cooling with nitrogen and ammonia to hydrogen combustion.

For the nitrogen results, Figure 4, zero blowing is seen to result in the highest plate temperatures. Increased flow rate of the subject coolant caused a progressive reduction in temperature at each location, with the greatest decrease noted at the rear station. These trends are consistent with established theory and experiment.

Temperature distributions from hydrogen tests with low blowing rates are seen to be higher than the zero blowing distribution, Figure 5. This is due to the combustion of the hydrogen-coolant. A very low blowing rate, F=0.00005, produced a maximum temperature at the 4.5 inches location. Increasing the blowing rate to F=0.00020 distributed the maximum temperatures reached over the 3.5 and 4.5 inches locations. With further increase in flow rate the temperature "high spots" moved towards the leading edge of the plate and for F=.0009 the two forward locations were at temperatures higher than they were for zero blowing. However, an increase to F>0.001 decreased the temperatures over the whole lower surface below the zero blowing values.

For the lowest blowing rates during the experiments with hydrogen, a thin blue flame sheet was visible along the plate surface, being thickest at the rear of the plate. At higher blowing rates the flame sheet thickened

along the entire length of the plate. Finally, it appeared to lift from the plate, slightly separated near the front and standing off approximately 0.25 inch at the rear of the plate. From these observations it seems that proximity of the visible flame to the plate surface resulted in high temperature zones on the plate (Figure 5). Flame size had no apparent correspondence to temperature level of the plate.

The results from the ammonia runs, Figure 6, indicate no strong effect of combustion or dissociation. The curves are similar in trend to the nitrogen curves. For $F>0.002$, using ammonia resulted in a greater temperature decrease than nitrogen achieved at the same blowing rate. A faint yellow flame was visible above the plate during the ammonia runs, and its behavior was similar to that of the hydrogen flame decribed above. The slightly higher temperatures at the rear locations for $F<0.0004$, Figure 6, are perhaps assignable to combustion effects.

For better comparison the results of the central four locations were reduced to a dimensionless temperature reduction factor:

$$TRF = (T_W - 70) \ / \ (T_{MS} - 70). \qquad (1)$$

The wall temperatures, T_W, were obtained by adding to the lower surface temperatures at each location the temperature difference across the wall, measured at the 1.5 inch location. Coolant inlet temperature of 70 °F was measured at the flow meter. The temperature reduction factors as a function of the blowing rates are plotted in Figures 7a and 7b.

From Equation (1) it can be seen that the temperature reduction factor becomes zero if the wall temperature attains the coolant inlet temperature (70 °F), and goes to one if the wall takes on the temperature of the

mainstream. Thus low values of TRF indicate effective cooling.

In each of the plots hydrogen use is seen to result in a reduced coolant effectiveness at low blowing rates; TRF for hydrogen increased due to combustion. As blowing rate increased, hydrogen became more effective (lower TRF) than either nitrogen or ammonia. The "crossover" point where hydrogen became more effective is seen to move to lower values of F as the rear of the plate is approached.

For the purpose of summarizing the experimental results the temperature reduction factors for the four locations discussed above were averaged and are presented in Figure 8. The effectiveness of hydrogen was reduced at low blowing rates ($F<0.001$) due to combustion. However, at higher blowing rates, the effectiveness of hydrogen was greater than the ammonia and nitrogen effectiveness. Combustion undoubtedly decreased hydrogen coolant effectiveness on all runs, but its effect is obvious only for those blowing rates which caused the plate temperature to increase above the level of zero blowing. A heat transfer analysis is necessary to evaluate the effect at higher blowing rates where temperature reduction was achieved.

2. Calculated Heat Transfer Results. A calculation of the convective heat transfer rate to the plate was made from a basic energy balance:

$$q_{conv} = q_{duct} + q_{base} + q_{coolant} \tag{2}$$

where q_{conv} is the convective heat transfer rate from the mainstream to the porous plate in Btu/hr-ft^2. The first term on the right represents the rate of radiation heat transfer from the plate to the surrounding duct in Btu/hr-ft^2 according to

$$q_{duct} = F\sigma(T_W^4 - T_D^4), \tag{3}$$

with F denoting the gray body shape factor from the plate surface (N-155, oxidized) to the stainless steel duct [11]. An estimate of F=0.3 was used for the computations. The calculated heat transfer rate by radiation from the upper to the lower radiation shield, Btu/hr-ft^2,

$$q_{base} = 0.9\sigma(T_{RS1}^4 - T_{RS2}^4), \tag{4}$$

was assumed to be equal to the rate of heat transfer loss through the cover plate of the plenum chamber because the coolant gas path did not include contact with the lower radiation shield. In the third term of Equation (2),

$$q_{coolant} = \frac{\dot{m}}{A} c_p(T_W - 70), \tag{5}$$

the convective heat transfer rate from the plate to the coolant, Btu/hr-ft^2, it was assumed that the coolant rose from its measured temperature, 70 °F at the flow meter, to the temperature of the upper wall surface, T_W, due to heat radiation from the lower surface and convection inside the porous wall.

Substituting Equations (3) with F=0.3, (4), and (5) into Equation (2) gives the basic energy balance:

$$q_{conv} = 0.3\sigma(T_W^4 - T_D^4) + 0.9\sigma(T_{RS1}^4 - T_{RS2}^4) + \frac{\dot{m}}{A} c_p(T_W - 70). \qquad (6)$$

For the presentation of the heat transfer results the two locations closest to the leading and trailing edges were not included. These locations were felt to be influenced by heat loss by conduction to the two ends. The remaining central four locations had equal shortest distances to the sides and their conduction losses were assumed to be negligible. The surface temperatures of the four locations were then averaged for all results shown here because each station had results similar to the average.

The convection heat transfer rates, computed from Equation (6), are shown in Figure 9. For each coolant gas it is seen that calculated heat transfer increased with increasing blowing rate. In all other studies of transpiration cooling which were encountered, heat transfer to the cooled surface was found to decrease with blowing rate increase. This conflict requires that the heat transfer calculation given above be questioned.

An examination for errors disclosed their possible occurence in the gray body shape factor for radiant heat exchange between upper surface and duct, or in the selection of the coolant entrance temperature and the coolant exit temperature. The Equations (3), (4), and (5), constituting the terms of the energy balance, were computed and plotted separately for comparison. These results are shown in Figures 10, 11, and 12 for hydrogen, ammonia, and nitrogen respectively. The two curves for $q_{coolant}$ (Equation (5)) were determined using the coolant inlet temperature measured at the flow

meter, 70 °F, (designated $q_{coolant,70}$), and the temperature values from measurements inside the plenum chamber, T_{CR}, ($q_{coolant,T_{CR}}$). The computations of Equation (3), the q_{duct} term, were made with the originally estimated value of F=0.3 and additionally with F=1.0, representing the maximum possible value under the condition of black body radiation.

As can be seen from Figures 10, 11, and 12, the results from the computations for ammonia and nitrogen were similar to those for hydrogen, but their coolant heat gains were smaller because of the lower value of specific heat for these gases. It is also shown that the difference between the curves $q_{coolant,70}$ and $q_{coolant,T_{CR}}$ is only minor. Thus, the choice of the coolant inlet temperature has only a slight effect on the results of Equation (6).

The radiation term is most important at low values of the blowing rate. In effect, the two different determinations of q_{duct} serve as bounds for the radiation heat transfer, with the upper curve, $q_{duct,F=1.0}$, prescribing an upper bound from black body analysis. The lower curve, $q_{duct,F=0.3}$, serves as a minimum bound determined from rather idealized material conditions.

The two resulting curves for the calculated heat transfer from mainstream to the plate, $q_{conv,F=1.0}$ and $q_{conv,F=0.3}$, differ only in the radiation term, q_{duct}, included in their determination (Equation (6)). Hence, they represent also bounding curves. For hydrogen and ammonia, (Figures 10 and 11), these bounding curves exhibit increasing trends with increasing blowing rates. Even though the lower bounds of q_{duct} showed a rate of decrease smaller than the upper bounds, the corresponding $q_{conv,F=0.3}$ curves resulted in the increase for both hydrogen and ammonia. This observation

leads to the exclusion of incorrectly determined gray body shape factor as the cause of heat transfer rate increase in question.

A sustaining argument for the exclusion made above can be found in the results from nitrogen runs, Figure 12. The lower radiation bound, $q_{duct,F=0.3}$, for nitrogen differed only slightly from the same curve for ammonia. For nitrogen, $q_{conv,F=0.3}$ is similar in trend to that seen for ammonia and hydrogen. However, the curve $q_{conv,F=1.0}$ for nitrogen decreased with blowing rate increase (where $q_{conv,F=1.0}$ for ammonia was noted to increase) although the values of $q_{duct,F=1.0}$, included in its determination, are higher than the ones for ammonia at same flow rates. From this it is seen that the radiation terms are not effecting the increase questioned above.

The rapid rate of increase of the $q_{coolant}$ term is seen to cause the total heat transfer rate, q_{conv}, to increase with increasing blowing rate. The slight decrease noted for nitrogen is primarily caused by its lower value of specific heat, resulting in a comparatively smaller rate of increase of $q_{coolant}$. Having eliminated coolant entrance temperature and gray body shape factor influences, the conclusion was reached that the coolant heat gain, $q_{coolant}$, is incorrectly determined. The reason for the high $q_{coolant}$ values is the unrealistic assumption that the coolant reached the wall temperature at the upper porous surface. This assumption may be valid for relatively thick walls of low porosity in systems where the wall temperature is only slightly (100 - 200 °F) above the coolant reservoir temperature. This type system is usually encountered in laboratory type studies of transpiration cooling using sintered powder porous materials.

In systems such as the one used in the present study, the wall thickness to pore size ratio is relatively small. Additionally, the coolant needs to experience a temperature rise of over 1000 °F in its passage through the 0.25 inch thick porous wall. That this can happen at the flow rates used seems improbable. This problem has been approached analytically by Shchukin and Koval'nogov [12], who showed that the relationship of coolant and wall temperature depends on the properties of the porous material.

The author feels that the method of heat transfer determination reported above is incorrect. A careful restudy, including analysis and experiment, of porous body cooling seems necessary before reliable heat transfer measurements can be made in systems of practical transpiration cooling.

PART IV

SUMMARY AND CONCLUSIONS

At low blowing rates, hydrogen combustion resulted in surface temperature increase above the zero blowing temperature level. However, it was found that, despite hydrogen combustion, hydrogen was more effective in reducing surface temperatures than ammonia and nitrogen at blowing rates of practical interest ($F>0.002$), where the coolant effectiveness was inverse proportional to the molecular weight, as predicted by analysis [2].

The combustion reaction is dependent on mainstream flow velocity and composition. The conditions of this experiment - low velocity (110 fps) and oxygen rich mainstream - favored the reaction. It can be assumed that the reduced effectiveness in high velocity, near stoichiometric mainstream flows will be less pronounced.

Because the method used for determining the total heat transfer rates was considered as incorrect, generalization of the results to other conditions is not possible. It seems that before accurate determination of heat transfer rates can be made for experimental systems of the practical type (wire mesh or perforated sheet porous sections and high heat transfer rates), the analysis of porous body cooling must carefully reexamined for material parameters and coolant flow rates of interest to transpiration cooling applications.

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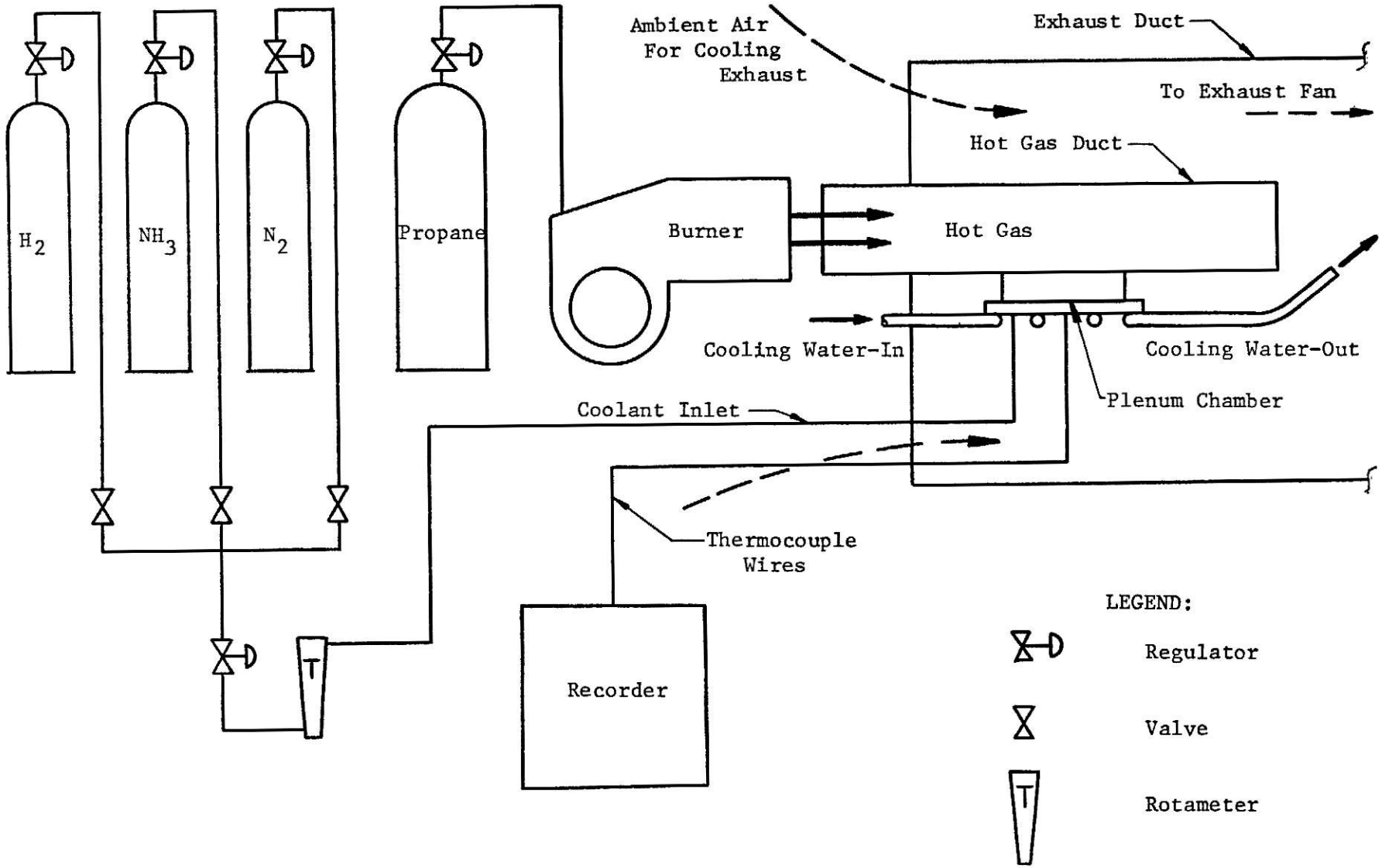

Figure 1 Experimental Apparatus

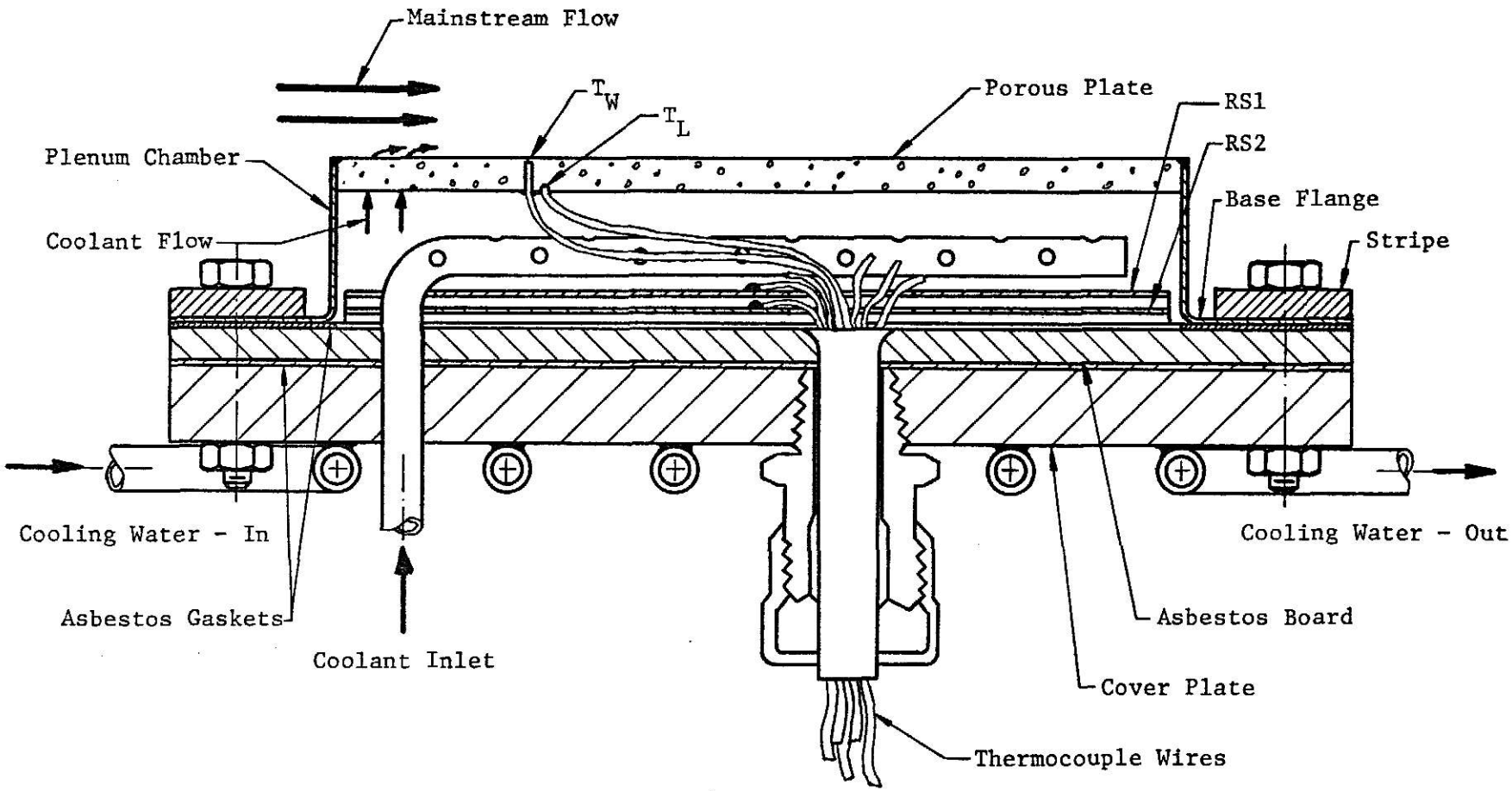

Figure 2 Porous Plate and Instrumentation

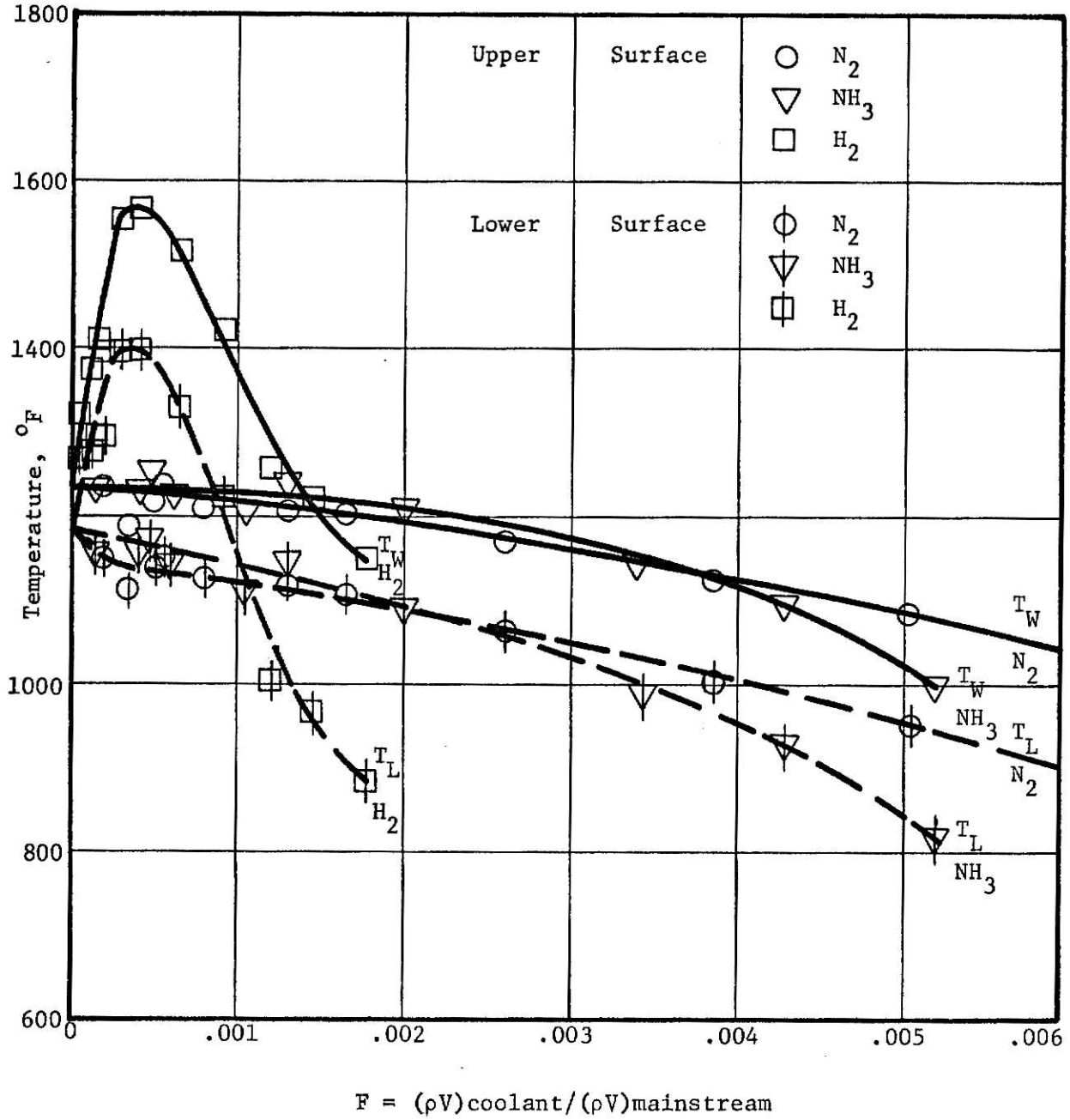

Figure 3 Temperatures of Upper and Lower Surfaces

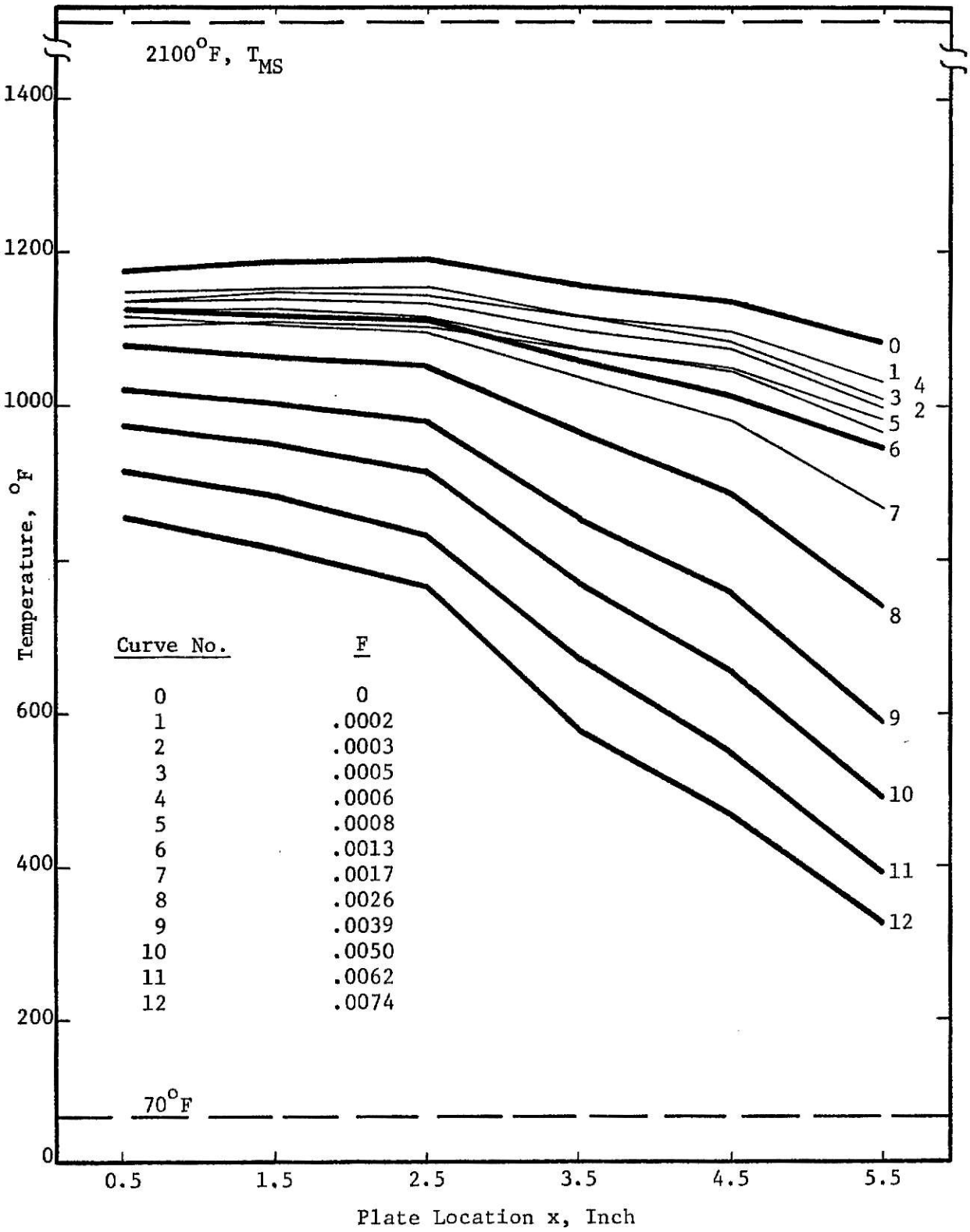

Figure 4 Lower Surface Temperatures: Nitrogen Runs

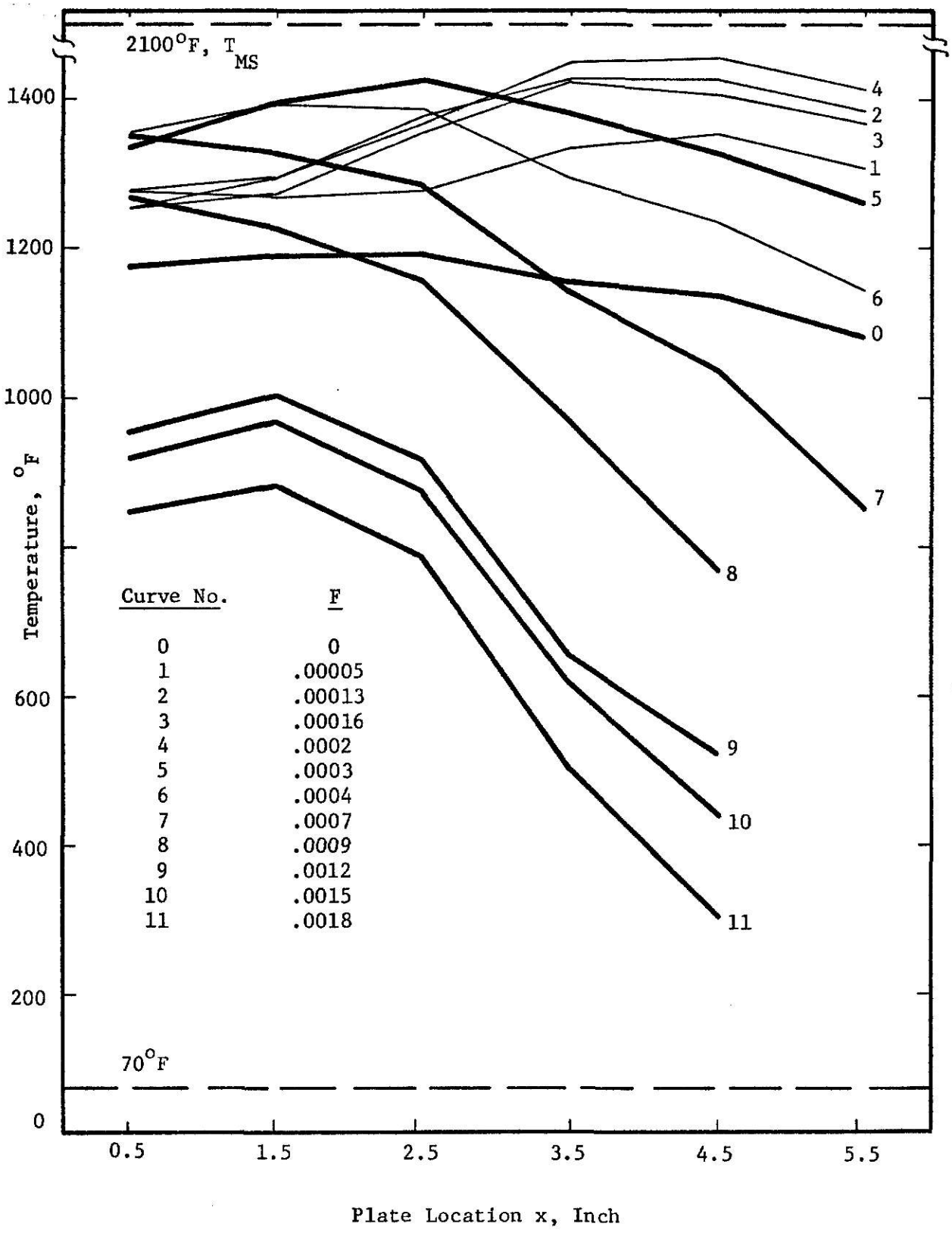

Figure 5 Lower Surface Temperatures: Hydrogen Runs

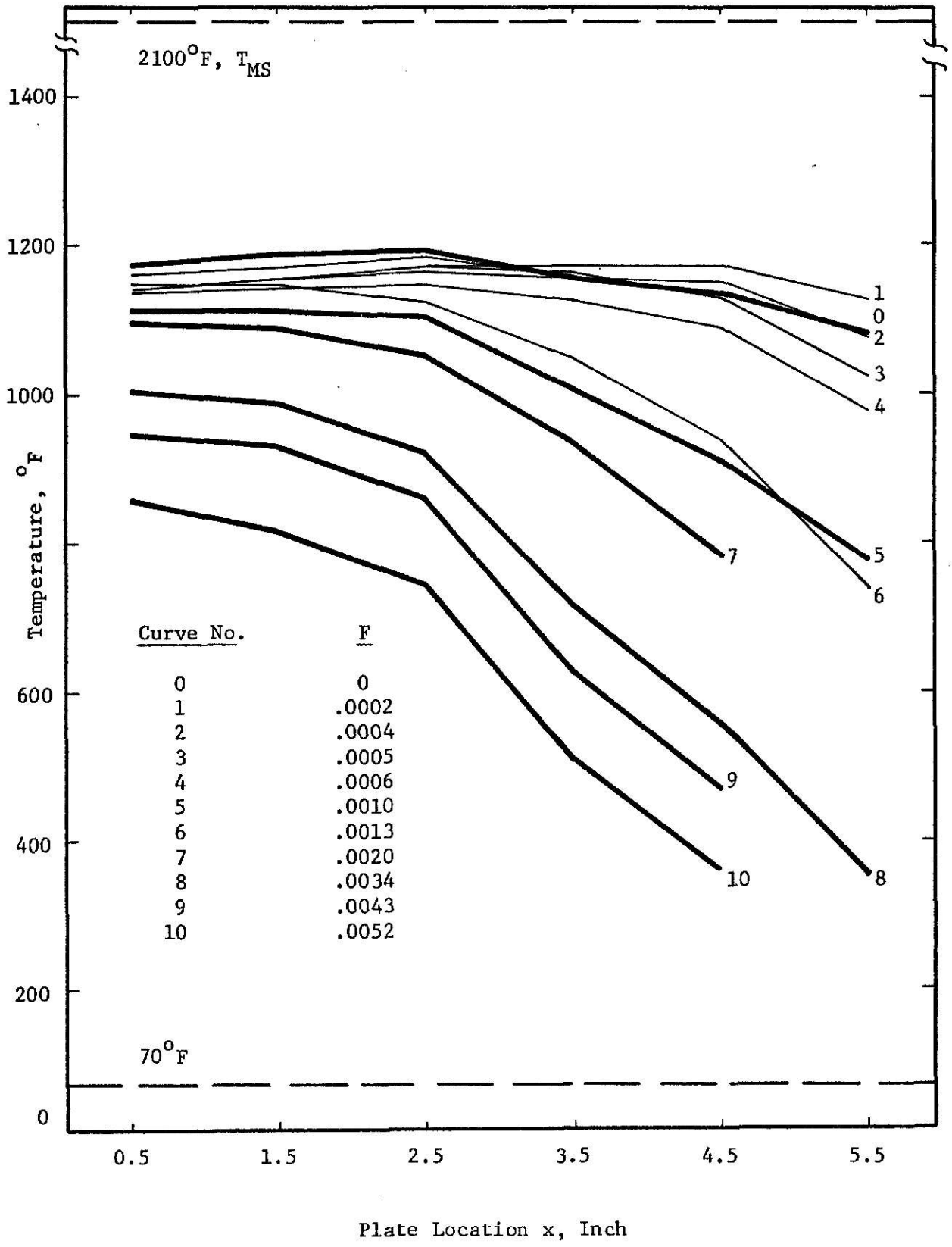

Figure 6 Lower Surface Temperatures: Ammonia Runs

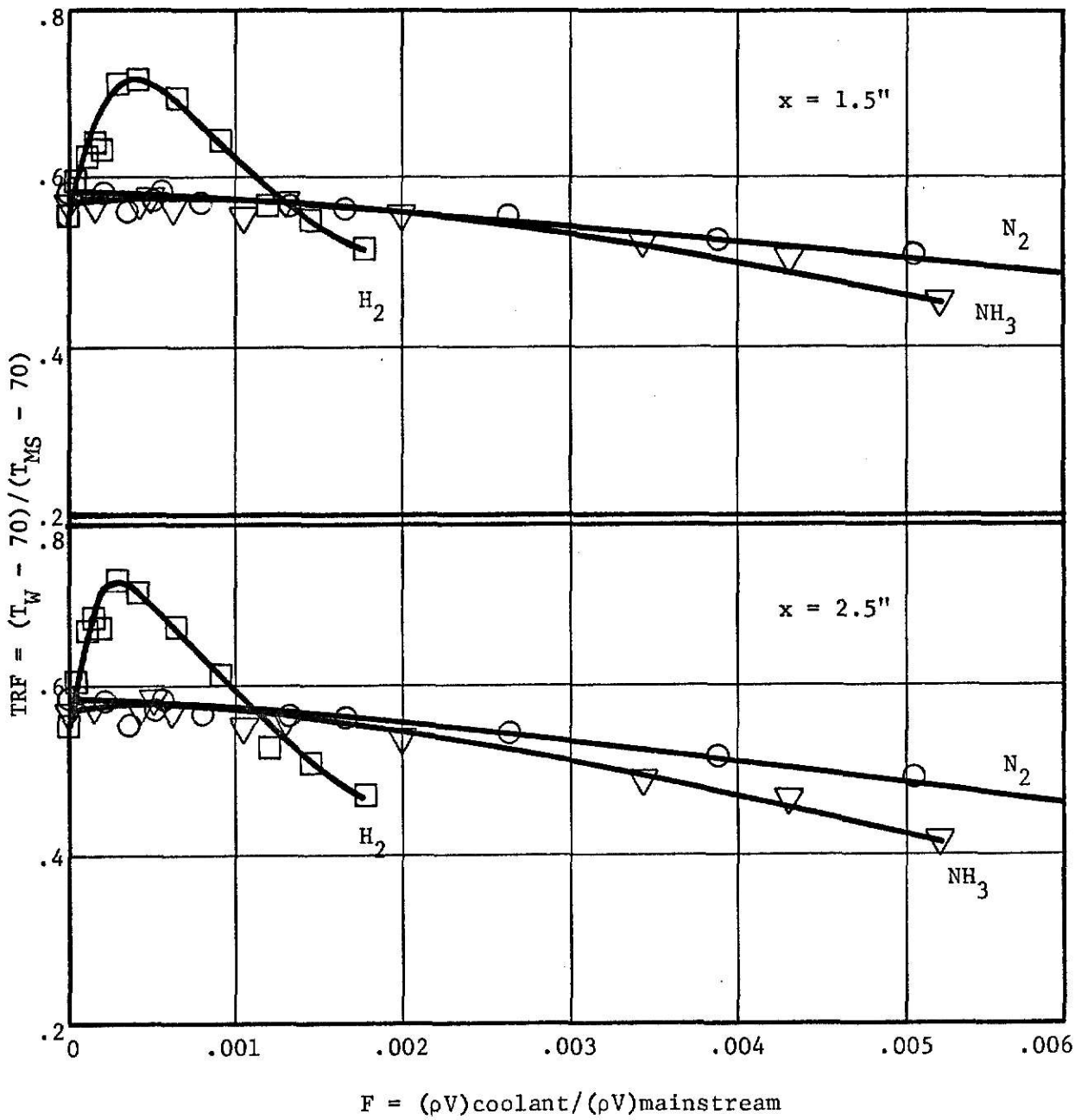

Figure 7a Temperature Reduction Factor at 1.5 and 2.5 Inch Plate Locations

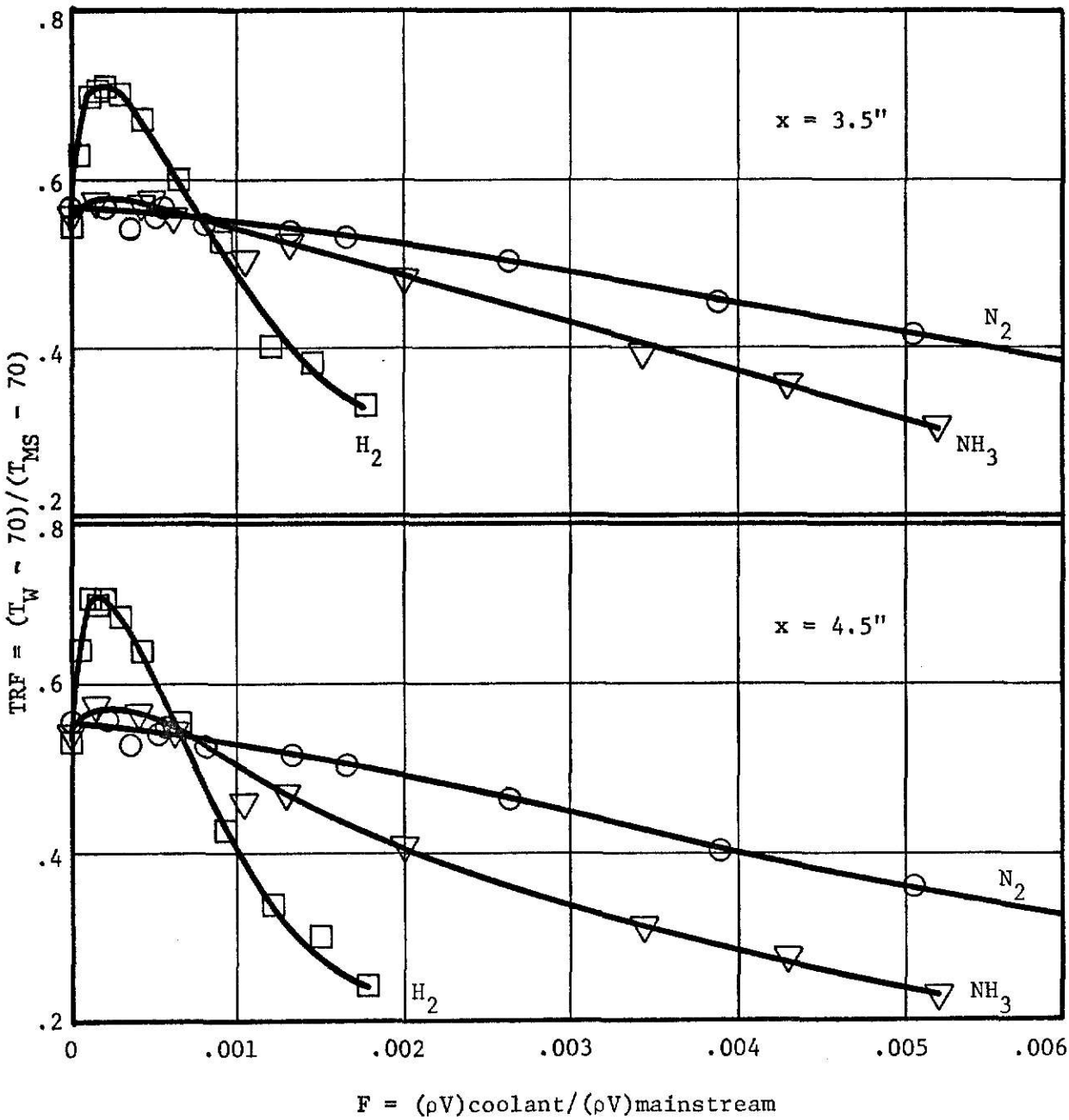

Figure 7b Temperature Reduction Factor at 3.5 and 4.5 Inch Plate Locations

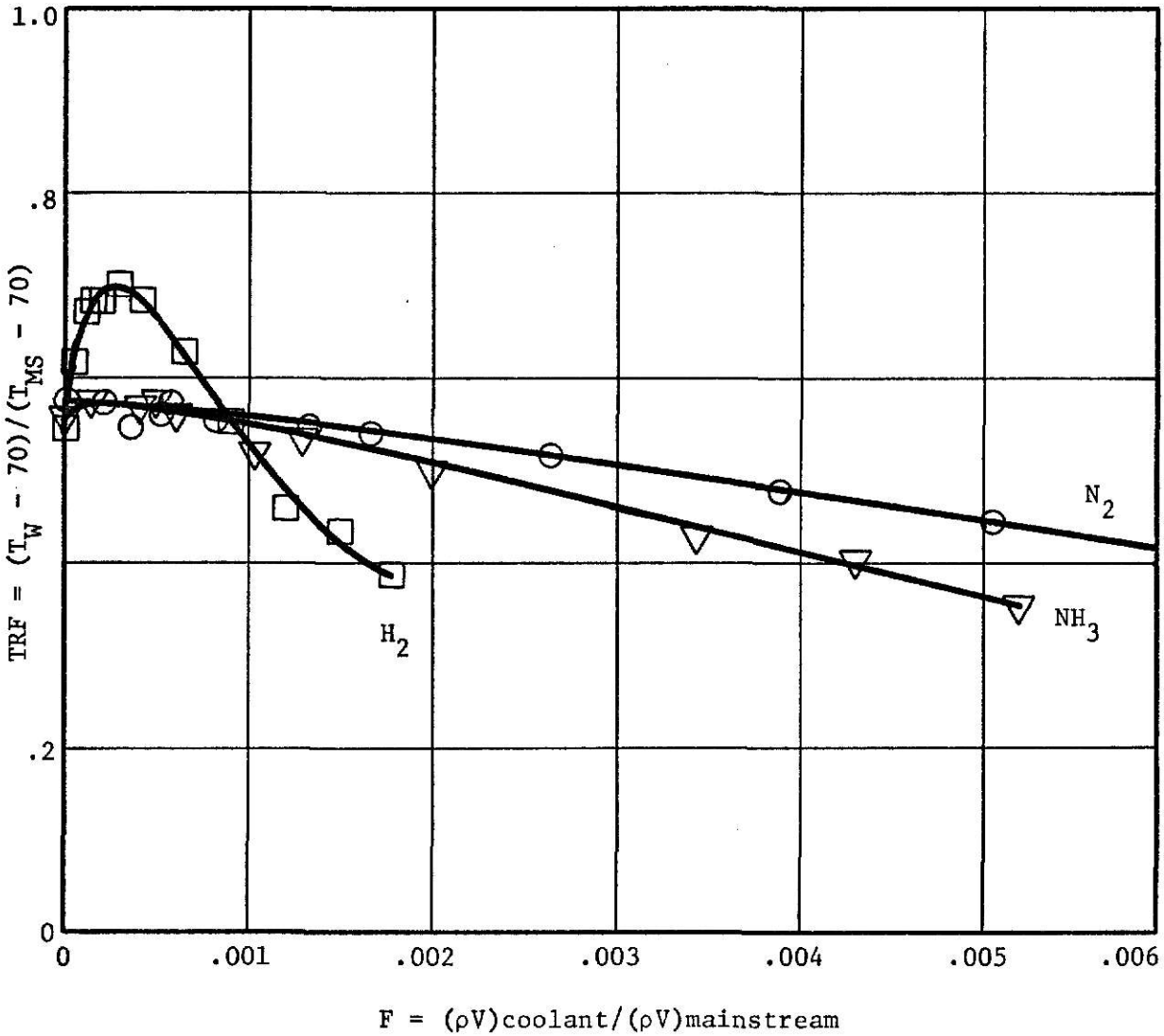

Figure 8 Temperature Reduction Factor: Average of 1.5, 2.5, 3.5, 4.5 Inch Plate Location Results

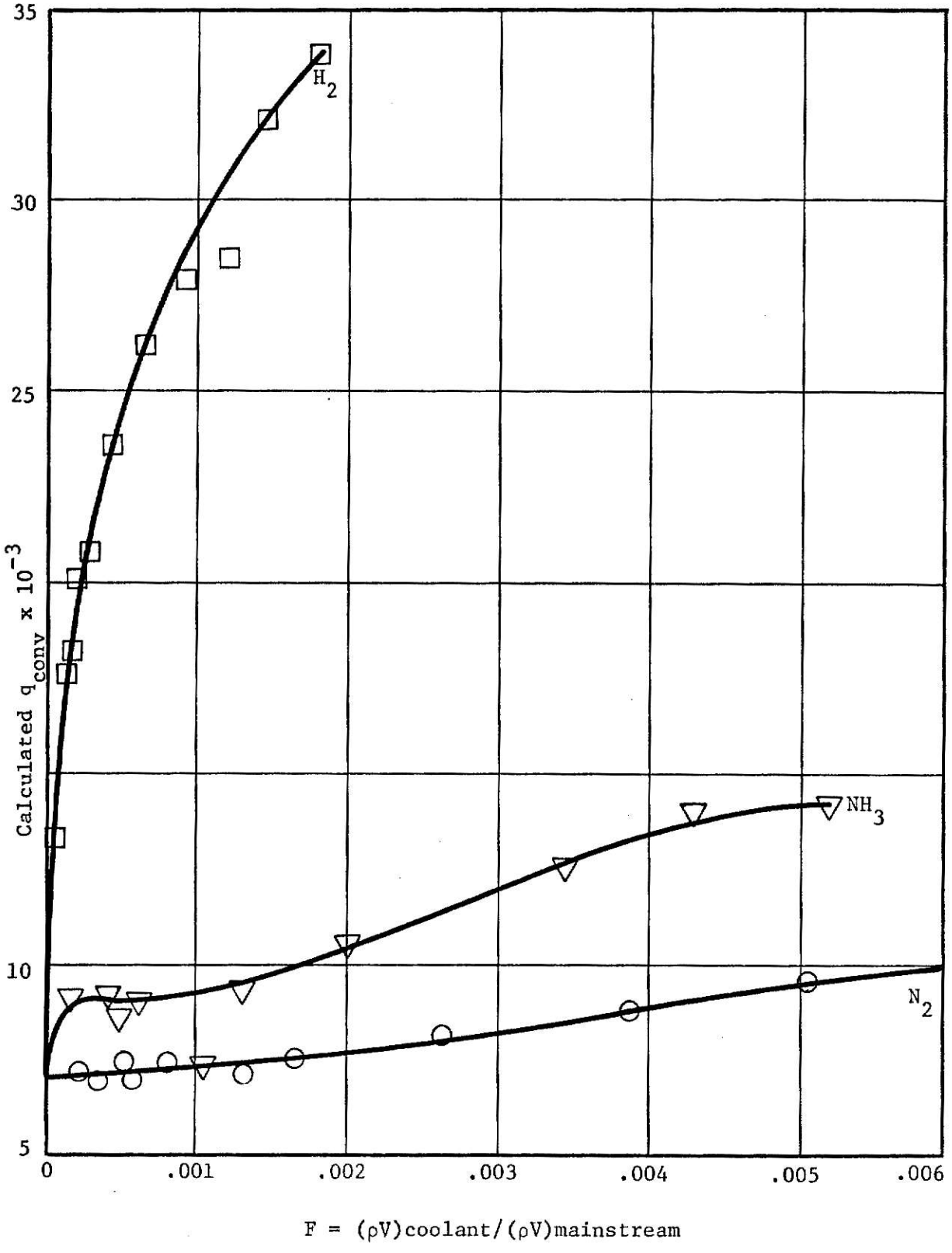

Figure 9 Calculated Net Heat Transfer

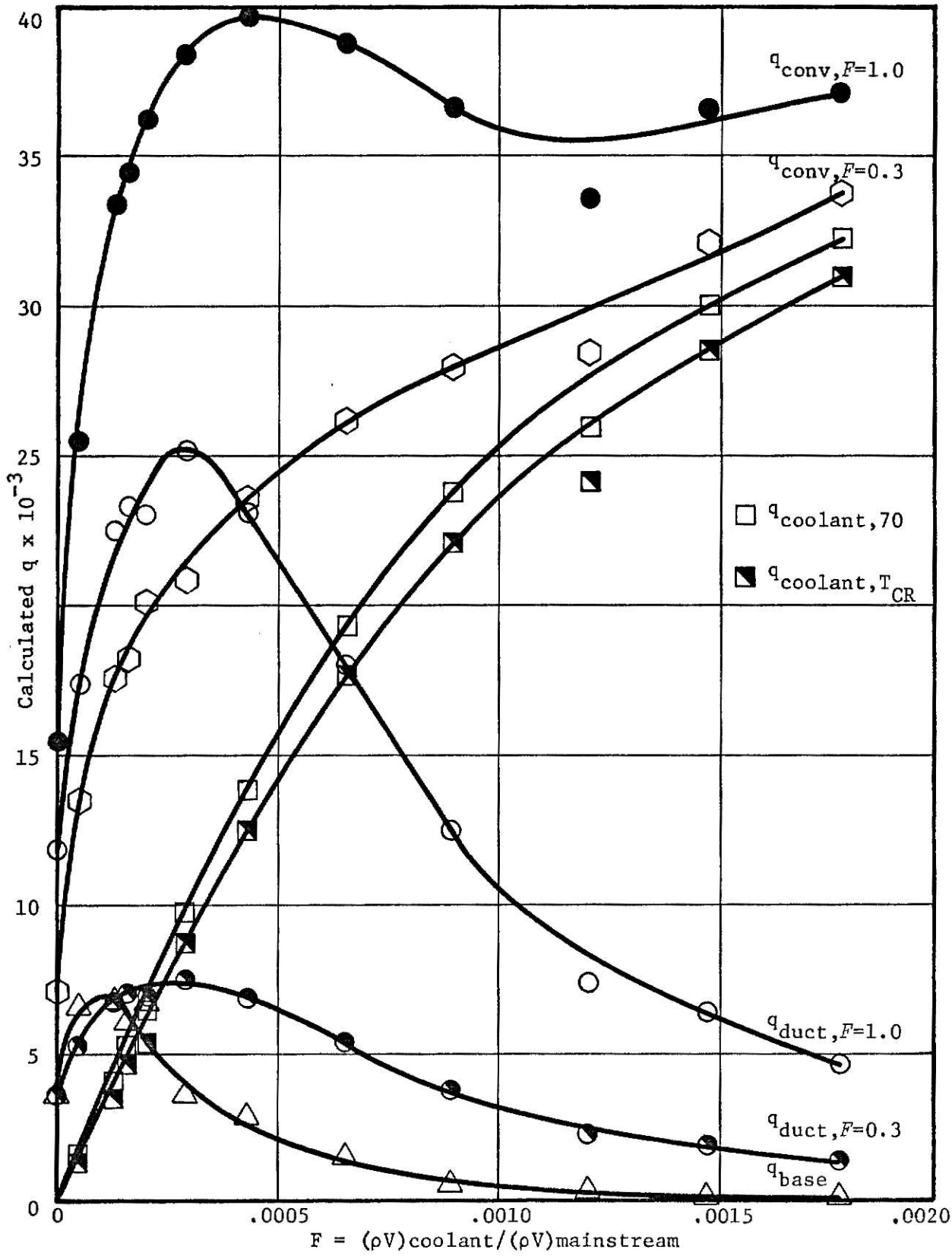

Figure 10 Detail Results of Heat Transfer Calculation: Hydrogen Runs

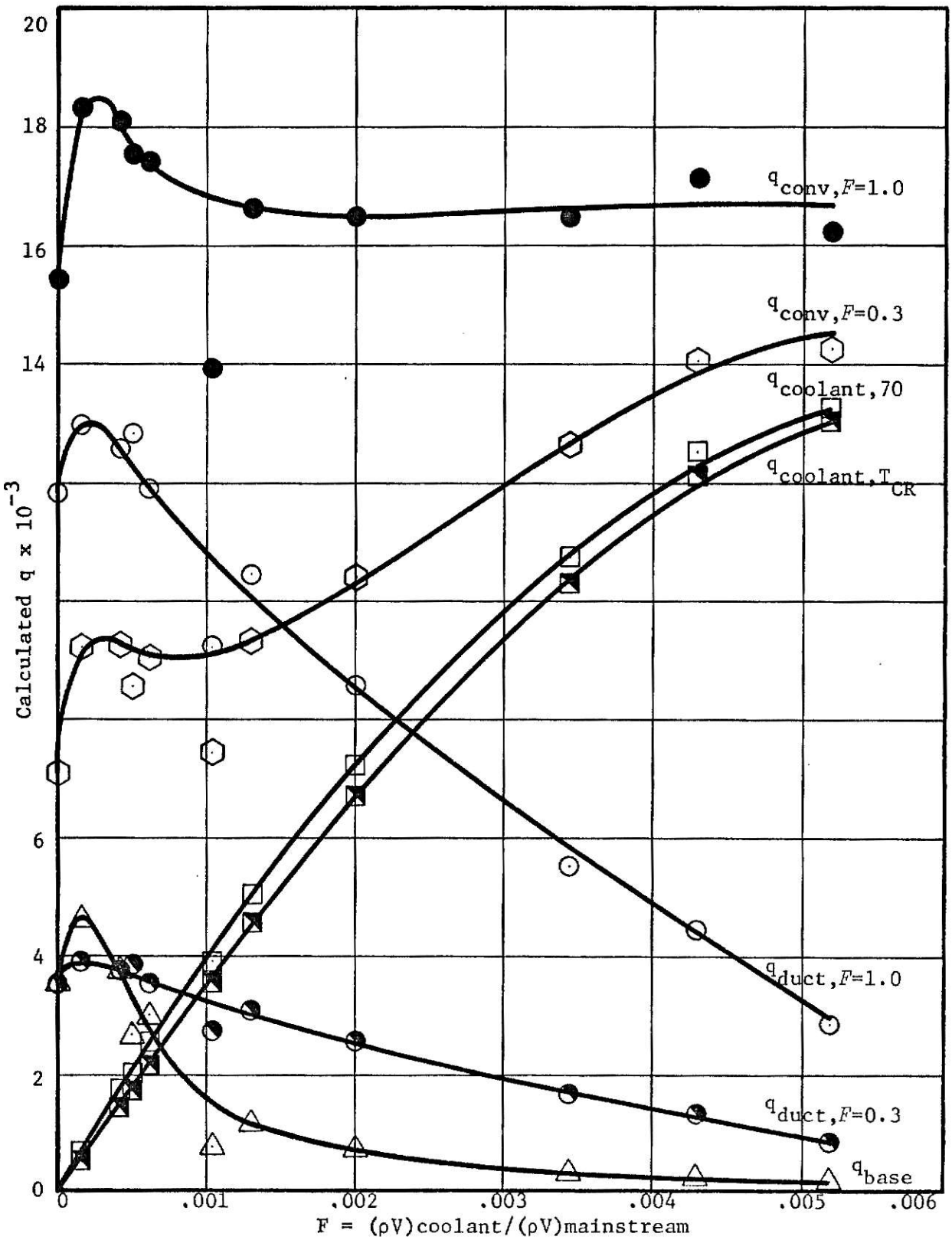

Figure 11 Detail Results of Heat Transfer Calculation: Ammonia Runs

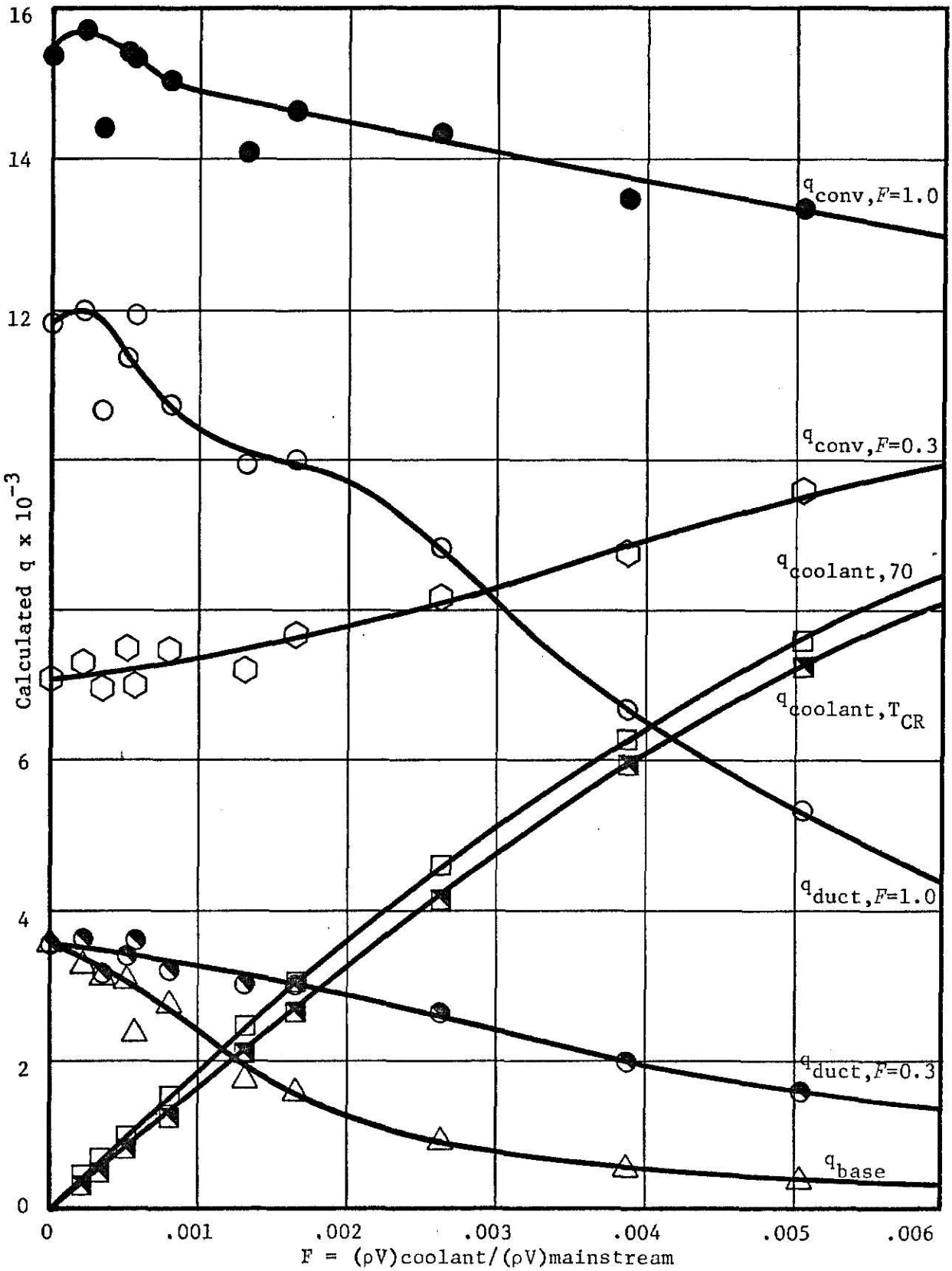

Figure 12 Detail Results of Heat Transfer Calculation: Nitrogen Runs

NOMENCLATURE

A	Porous surface area, ft^2
c_p	Coolant specific heat, Btu/lb-^oR
F	Blowing rate parameter, $F = (\rho V)coolant/(\rho V)mainstream$
$\dot{m}$	Coolant mass flow rate, lb/hr
T_{CR}	Coolant reservoir temperature, ^oR
T_D	Duct temperature, ^oR
T_L	Porous solid temperature, lower surface, ^oR
T_{MS}	Mainstream gas temperature, ^oR
T_{RS1}	Upper radiation shield temperature, ^oR
T_{RS2}	Lower radiation shield temperature, ^oR
T_W	Porous solid wall temperature, upper surface, ^oR
V	Velocity, ft/sec
$\mathcal{F}$	Gray body shape factor, [11]
σ	Stephan-Boltzmann constant, Btu/hr-ft^2-^oR
ρ	Density, lb/ft^3

ACKNOWLEDGEMENTS

I would like to acknowledge my indebtedness to Dr. R. L. Gorton for his valuable assistance during the building of the experimental apparatus and the performance of the experiments. His excellent guidance in the preparation of this thesis is gratefully appreciated.

I am thankful for the support of this study by the National Science Foundation (NSF Grant GK-1557) and the Kansas State University Engineering Experiment Station.

Thanks are also due to Mr. A. W. Brecheisen II who was responsible for preliminary work which led to the present study.

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AN EXPERIMENTAL STUDY OF COOLANT COMBUSTION EFFECTS IN TRANSPIRATION COOLING

by

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und Hansestadt Hamburg, 1964

AN ABSTRACT OF A THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

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1970

ABSTRACT

The protection of structures from high convective heating can be accomplished by application of mass transfer cooling in three fundamentally different ways, of which transpiration cooling is the one with greatest effectiveness. This is attained due to the heat sink properties of the coolant passing through the porous structure or porous wall. Additionally, after the coolant emerges from the porous wall into the boundary layer it forms a protective layer, reducing heat transfer to the wall.

Practical applications of transpiration cooling have been executed with air as coolant for gas turbine blades and vanes, exposed to moderate temperatures. For cooling structural parts under exposure of higher temperatures (above 2000°F) the demand is for a more effective coolant. Analysis of porous body (transpiration) cooling has shown that the effectiveness of coolants increases as their molecular weight decreases. This places the fuel, used in many systems exposed to high convective heating, among the obvious alternate coolants. An advantage is the availability of the fuel energy after the fuel has entered the mainstream flow. However, combustion of the fuel-coolant could result in the adverse affect of increased surface temperatures rather than a reduction.

To provide preliminary information on the influence of boundary combustion on transpiration cooling effectiveness, this study was undertaken. It was limited to a single surface (a flat plate of woven wire mesh) under one set of heating gas flow condition. The coolants chosen were hydrogen and ammonia because of their reactive potentials (hydrogen appears also as a

fuel in some of the proposed applications), and nitrogen as a non-reactive comparison coolant.

Varying coolant flow rates of the three gases were injected through the porous plate, whose upper surface was positioned in the mainstream flow. Thermocouples were installed at several centerline locations on the lower plate surface and one on the upper surface to obtain temperature data and to determine heat transfer rates. Temperature measurements for zero blowing (no coolant flow) were made for comparison purposes.

The temperature difference across the porous plate was noted to increase slightly with blowing rate ($F = (\rho V)_{coolant}/(\rho V)_{mainstream}$).

For the nitrogen runs, plate temperatures were found to gradually decrease with increasing blowing rates. Similar results were obtained with ammonia cooling. For hydrogen runs, low blowing rates resulted in plate temperatures higher than the zero blowing temperature distribution. This was attributed to hydrogen combustion. With increased blowing rates a considerable temperature reduction was achieved.

A temperature reduction factor (TRF) indicating cooling effectiveness was computed separately for four central plate locations and for their average. The plots of temperature reduction factor versus blowing rate showed, at higher blowing rates ($F > 0.002$), hydrogen was more effective in surface temperature reduction than the other two coolants. The effectiveness was inverse proportional to coolant molecular weight, supporting analytical predictions.

To evaluate the effect of combustion at higher blowing rates where a temperature reduction was achieved, the convective heat transfer rates were calculated, applying a basic energy balance. The results obtained from this

computation showed for all three coolants trends which were opposite to those found in the literature encountered. This led to an investigation of the calculations and assumptions made.

It is demonstrated that assumption of temperature equality of coolant and wall leads to erroneous results in heat transfer determination.