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Measurements of Convective Heat-Transfer
Coefficients for the Subsonic Flow of
Hydrogen and Helium in Smooth Tubes at
Moderate Surface and Fluid Temperatures

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Measurements of Convective Heat-Transfer Coefficients for the Subsonic Flow of Hydrogen and Helium in Smooth Tubes at Moderate Surface and Fluid Temperatures

by

J. A. Sullivan
C. R. King

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NOMENCLATURE

- A_c = test-section cross-sectional area, cm^2
 A_1 = flow cross-sectional area of test section, cm^2
 C_p = coolant constant-pressure specific heat, cal/g K
 D = inside diameter of test section, cm
 Δe = voltage drop across a thermocouple due to test-section voltage gradient, V
 F_{w-s} = radiation shape factor for radiative heat exchange between concentric cylinders (see Eq.(4).)
 h_l = local heat-transfer coefficient, $\text{cal/sec cm}^2 \text{ K}$
 h_x = static enthalpy of coolant at axial Position x , cal/g
 I = test-section current, A
 k_x = test-section thermal conductivity at Location x , cal/sec cm K
 Nu_b = Nusselt number calculated by using the local heat-transfer coefficient of Eq. (9) and coolant properties evaluated at T_b
 Pr_b = Prandtl number calculated by using T_b to evaluate coolant properties
 q''' = volumetric heat generation in test-section wall, cal/sec cm^3
 Q_f = heat flow to coolant, cal/sec
 q_f'' = heat flux to coolant, cal/sec cm^2
 q_{fl}'' = local radial heat flux to coolant, cal/sec cm^2
 q_r'' = radiative heat flux from test-section, cal/sec cm^2
 q_x'' = longitudinal heat flux in test-section wall at Location x , cal/sec cm^2
 Re_b = Reynolds number based on coolant flow and local fluid properties evaluated at T_b ; ($Re_b = wD/\mu_b A_1$)
 r_i = test-section inside radius, cm
 r_o = test-section outside radius, cm
 r_s = inside radius of first radiation shield, cm
 T_b = local static coolant temperature, K
 T_{f1} = coolant static temperature at inlet, K
 T_{o1} = coolant stagnation temperature at inlet, K
 $\bar{T}_s$ = average radiation-shield temperature for an axial Increment Δx , K
 $\bar{T}_w$ = average test-section wall temperature for an axial Increment Δx , K
 ΔV = test-section voltage drop over an Increment Δx , V
 V_x = coolant axial velocity at axial Position x , cm/sec

w = coolant flow rate, g/sec

Δx = axial increment, cm

x = axial distance from start of heating, cm

ϵ_s = emissivity of graphite radiation shield

ϵ_w = emissivity of test-section wall

ρ_{f1} = coolant inlet density, g/cm³

ρ_m = resistivity of molybdenum, Ω cm

σ = Stephan-Boltzmann constant

μ = coolant viscosity, g/sec cm

MEASUREMENTS OF CONVECTIVE HEAT-TRANSFER COEFFICIENTS FOR THE SUBSONIC FLOW OF
HYDROGEN AND HELIUM IN SMOOTH TUBES AT MODERATE SURFACE AND FLUID TEMPERATURES

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ABSTRACT

Experimental investigations into the forced-convection heat transfer for helium and hydrogen flowing in smooth tubes were conducted at a nominal inlet temperature of 300 K, wall temperatures to 1475 K, exit-gas temperatures to 725 K, inlet Reynolds numbers from 10^4 to 7×10^4 , and heat fluxes to $31.4 \text{ cal/sec cm}^2$. The length-to-diameter ratio of the test section was 87, and maximum wall-to-bulk temperature ratios in the experiments reached 1.9. The results agree well with a recently published correlation of experimental data, but preliminary comparisons of these data with a two-dimensional numerical solution for turbulent flow in a heated tube show only moderate agreement.

INTRODUCTION

The technology associated with nuclear-fueled, hydrogen-cooled rocket engines has made it important to extend convective-heat-transfer investigations to very high wall temperatures ($> 2800 \text{ K}$) and high wall-to-bulk temperature ratios (> 20), because at such high ratios, large variations in fluid properties transverse to the flow direction occur, making heat-transfer correlations based on conventional dimensionless parameters no longer valid. Several experimental investigations (see bibliography of Ref. 1) wherein transverse property variations are important have been directed toward obtaining heat-transfer data and toward a suitable correlation of these data. However, most correlations are not general, and only that of M. F. Taylor (1) is reasonably successful for single-phase heat transfer to hydrogen and other gases except near the critical point and near the inlet region. Analytical investigations of the problem have been conducted, notably by R. G. Deissler (2); Sparrow, Hallman, and Siegel (3); P. M. Magee (4); D. M. McEligot, S. B. Smith, and C. A. Bankston (5); and by C. A. Bankston and D. M.

McEligot (6).

The results presented herein are compared to the numerical solution of the coupled, partial-differential boundary layer equations, using the code developed by C. A. Bankston and D. M. McEligot (6,7) and to the most recent correlation of M. F. Taylor (1). While the experimental results do not extend to high wall-to-bulk temperature ratios (T_w/T_b), as was originally planned, they do represent consistent data up to $T_w/T_b = 1.9$ and afford an interesting comparison with the work of Bankston and McEligot for the case of non-constant wall heat flux. Because of the premature curtailment of this experimental program for monetary reasons, the study is incomplete and more work is necessary, particularly with regard to comparisons with available analytical solutions.

APPARATUS AND PROCEDURE

Facility

The hydrogen and helium heat-transfer experiments were conducted at the hazardous-gas test facilities at Los Alamos Scientific Laboratory's TA-46

site. A high-pressure gaseous-flow system, which was remotely controlled from outside the experimental area, supplied either gaseous helium or gaseous hydrogen to an electrically heated test section. All instrumentation and control circuitry was passed from the experimental area through a concrete wall to a control room where all data-recording and control functions were performed.

An overall view of a portion of the facility including the heat-transfer apparatus is presented in Fig. 1. A schematic diagram of the gas-flow system is shown in Fig. 2. Either hydrogen or helium gas was supplied to the test section through a pressure-regulating valve, which was also used to control the flow rate of the particular gas through the test section. The flow rate was determined with a calibrated critical-nozzle assembly before the gas flowed into a counter-flow mixing chamber at the entrance to the test section. Here the incoming gas was thoroughly mixed to ensure accurate measurement of the inlet

bulk gas temperature by a copper/constantan thermocouple at the exit of the mixing chamber. After passing through the test section, the gas entered another mixing chamber where a platinum-platinum-10% rhodium thermocouple was used to measure the exit bulk gas temperature.

A water-cooled heat exchanger was located in the pass-through at the top of the furnace shell to prevent the hot gas from contacting the shell and to provide a good gas seal. An O-ring seal allowed for differential thermal expansion between the warm furnace parts and the relatively cold furnace pressure shell. A second water-cooled heat exchanger reduced the gas temperature to ambient before passing the gas through the backpressure-regulating valve. After passing through the backpressure regulator, the gas was discharged into a vent system which carried the gas up an exhaust stack.

Bypass electrical conductors around the inlet and exit mixing chambers (Fig. 2) reduced the voltage drop across the stainless-steel mixing chambers.

Fig. 1. Overall view of heat-transfer test facility.

Fig. 2. Schematic diagram of test apparatus and gas flow system.

These conductors were fabricated from 19.05-mm-wide braided-copper ground strapping held in place on both ends by a simple hose clamp. By adding these straps, we could generate higher power in the test section for a given power input, thereby providing for more efficient usage of the power supply. Also, the straps reduced the addition of unwanted heat to the inlet and exit mixing chambers. Power was supplied to the test section by a Hewlett Packard dc power supply capable of delivering 1000 A at 8 V with a very low ripple. The power input to the furnace was monitored by a simple 50-mV/1000-A shunt.

The two vacuum pumps shown in Fig. 2 were used to obtain and maintain a vacuum inside the furnace pressure vessel. This vacuum was maintained (1) to prevent air from contacting the molybdenum test section and (2) to minimize the heat-conduction losses from the test section during testing.

Eight voltage taps fabricated from 0.254-mm-diam molybdenum wire were located on the test section (Fig. 2). The end taps were positioned as close as possible to the flanges to minimize the extrapolations necessary for obtaining the voltage drop across the test section. Two pressure taps were located in the flanges on both ends of the test section. Both taps were brazed to the flanges and communicated with the test gas in the tube through holes which had been formed by the electrical discharge machining (EDM) technique with a 0.431-mm-diam electrode. The resultant diameter of these pressure ports was 0.457 mm. (EDM is extremely useful for the machining of pressure ports because the backward arcing during electrode breakthrough ensures the absence of burrs around the inside edges of the hole.)

Test Section

A detailed drawing of the test section, radiation shields, and inlet and exit mixing chambers is presented in Fig. 3. The locations of the thermocouples and pressure taps are indicated in centimeters although the original design used the British system of units. The dimensions indicated in Fig. 3 are nominal; actual dimensions obtained with a precision cathetometer for both test sections are listed in Tables I and II. These dimensions were measured after all the data had been obtained. Short pieces were cut from the ends and from the center of each test section to determine their inside and outside diameters and the roughness of the bore surfaces. There was little variation from section to section, and the arithmetic average of each of the three sets of readings was therefore used in the data-reduction calculations for a given test section. Test Section 1 failed early in the experimental program during a high-power low-flow run. Thus most of the results were obtained with Test Section 2, which performed well for the remainder of the experimental program. As is evident from Fig. 3, the thermocouples were positioned closely at both ends of the test section to ensure greater accuracy in determining the end heat-conduction losses.

A sliding joint at the lower end of the test section (see Fig. 2) was designed to accommodate its thermal expansion of the test section while maintaining a good gas seal. The top flange of the section was bolted to the mounting platform, and a metallic O-ring was used as a gas seal at this interface.

At each axial location, the local wall temperature was measured with two thermocouples spaced azi-

Fig. 3. Test-section geometry.

TABLE I
INSTRUMENTATION DATA FOR TEST SECTION 1

Length, cm 32.275
Outside diameter, cm 0.4763
Inside diameter, cm 0.3711
Surface roughness, cm-rms 56×10^{-6}
Pressure-tap distance, cm 33.705

Right-Side Thermocouple		Left-Side Thermocouple		Voltage Tap	
Number	Location, cm	Number	Location, cm	Number	Location, cm
1	0.335	1	0.335	1	0.135
2	1.405	2	1.420	2	3.995
3	3.518	3	3.457	3	8.656
4	7.645	4	7.607	4	13.536
5	12.682	5	12.670	5	18.738
6	17.805	6	17.762	6	24.000
7	22.924	7	22.893	7	28.344
8	27.341	8	27.305	8	32.111
9	30.701	9	30.655	-	--
10	32.083	10	32.070	-	--

Radiation shield diameter, cm 4.125
First shield thermocouple, cm 1.422
Second shield thermocouple, cm 16.027
Third shield thermocouple, cm 31.267

TABLE II
INSTRUMENTATION DATA FOR TEST SECTION 2

Length, cm 32.347
Outside diameter, cm 0.4750
Inside diameter, cm 0.3721
Surface roughness, cm-rms 53×10^{-6}
Pressure-tap distance, cm 33.788

Right-Side Thermocouple		Left-Side Thermocouple		Voltage Tap	
Number	Location, cm	Number	Location, cm	Number	Location, cm
1	0.360	1	0.304	1	0.151
2	1.459	2	1.466	2	4.241
3	3.520	3	3.502	3	8.706
4	7.615	4	7.460	4	13.818
5	12.757	5	12.721	5	19.090
6	17.842	6	17.852	6	23.780
7	22.908	7	22.897	7	28.524
8	27.325	8	27.298	8	32.215
9	30.690	9	30.648	-	--
10	32.051	10	32.051	-	--

Radiation shield diameter, cm 4.125
First shield thermocouple, cm 1.422
Second shield thermocouple, cm 16.027
Third shield thermocouple, cm 31.267

muthally 120° apart (Fig. 4). The thermocouple wires passed halfway around the test section before being threaded through ceramic insulators in the radiation shields. To minimize conduction heat losses, 0.254-mm-diam platinum-platinum-10% rhodium thermocouple wire was used in this installation. All wall thermocouples and voltage taps were attached to the test section with a capacitor discharge welding machine. The physical arrangement of the test section with the radiation shields in place and all instrumentation connected is shown in Fig. 5.

The entire flow system was thoroughly purged with helium gas at the beginning of each series of tests to remove all traces of air from the flow system. Any remaining air coming in contact with the hot molybdenum test section would have shortened its useful life considerably. After the purge cycle, a fairly high flow rate of either helium or hydrogen gas was set and the power supply was turned on to begin heating the test section. After a selected wall temperature was reached, the flow rate was reduced to the desired value. Experimental data were obtained for both gases at constant power inputs by varying the flow rates. Each power-input sequence for both gases was repeated until all desired data were obtained before the power input was raised to a new level.

Most data were recorded on two different recording systems. Minneapolis Honeywell Brown strip-chart recorders were used to continuously monitor

the important data channels and to indicate any trends during testing. Simultaneously, a fairly large portion of the instrumentation was monitored by a Hewlett Packard Dymec 2010 D digital-data acquisition system, and data were recorded by two on-line output devices. IBM 8-level perforated paper tape was used for future computer processing. The same data were simultaneously supplied to a Hewlett Packard 2114A computer, which provided rapid on-line data reduction of parameters needed for monitoring the stability of each data point (e.g., electrical power input, gas flow rate, exit-gas temperature, and test-section wall temperatures). The data-taking process was thus carefully controlled, with a minimum of data points being recorded.

Fig. 4. Thermocouple installation geometry.

Fig. 5. Closeup view of test section with radiation shields in place.

DATA REDUCTION TECHNIQUE

Average local values of heat transfer to the coolant gas were obtained by dividing the test section into 20 unequal axial segments plus two very small increments at the first and last wall thermocouple positions. The two small increments were used to obtain a smooth transition from the measured portion of the wall-temperature profile to the measured portion. Axial increments were chosen so as to minimize error due to averaging over locations where the temperature profile was varying rapidly. This technique was made possible by using a data-smoothing splinefit to obtain a smooth axial temperature profile which passed through measured wall temperatures. A linear profile was used at the inlet and exit ends of the test section, with the measured fluid temperatures serving as end points.

An arbitrary axial increment together with the attendant heat flow is shown schematically in Fig. 6.

A heat balance on the element shown gives

$$\begin{aligned} (q_x'' - q_{x+\Delta x}'') \pi(r_o^2 - r_i^2) - q_x'' 2\pi r_o \Delta x - q_y'' 2\pi r_i \Delta x \\ + q'' \pi(r_o^2 - r_i^2) \Delta x = 0. \end{aligned} \quad (1)$$

The heat flow in the axial direction is determined from

$$\left. \begin{aligned} q_x'' &= -k_x \left(\frac{dT_w}{dx} \right)_x \\ q_{x+\Delta x}'' &= -k_{x+\Delta x} \left(\frac{dT_w}{dx} \right)_{x+\Delta x} \end{aligned} \right\}, \quad (2)$$

Fig. 6. Arbitrary test-section wall element.

where the conductivity of molybdenum was evaluated at the outer-wall temperatures at x and $(x+\Delta x)$ and the temperature profile derivatives were available from the computer fit to the temperature profile. Data for the conductivity of molybdenum used in the analysis are presented in Fig. 7. The radiation heat exchange between the test section and the radiation shield was calculated by using the expression

$$q_r'' = F_{w \rightarrow s} \sigma (\bar{T}_w^4 - \bar{T}_s^4), \quad (3)$$

where

$$F_{w \rightarrow s} = \frac{1}{\frac{1}{\epsilon_w} + \frac{r_o}{r_s} \left(\frac{1}{\epsilon_s} - 1 \right)}. \quad (4)$$

Numerical values for ϵ_w were evaluated at the average wall temperature for an increment, and the emissivity of the graphite radiation shield was assumed constant at 0.8. Values of ϵ_w used in the analysis are shown in Fig. 8. Both average wall and average shield temperatures are used in Eq. (3).

The power input to an axial increment is given by

$$q'' = I \Delta V, \quad (5)$$

where I is the measured test-section current and ΔV is the voltage drop across the increment. Values of ΔV were obtained by fitting a smooth curve

Fig. 7. Conductivity of molybdenum as a function of temperature.

Fig. 8. Emissivity of molybdenum as a function of temperature.

through the measured test-section voltage profile with the spline routine used to fit the temperature profile.

Combining Eqs. (1), (2), (3), (4), and (5) and solving for the heat flow to the coolant gives

$$Q_f = q_f'' 2\pi r_1 \Delta x = \left[k_{x+\Delta x} \left(\frac{dT_w}{dx} \right)_{x+\Delta x} - k_x \left(\frac{dT_w}{dx} \right)_x \right]$$

$$\pi(r_0^2 - r_1^2) - \frac{2\pi r_0 \Delta x \sigma}{\frac{1}{\epsilon_0} + \frac{r_0}{r_s} \left(\frac{1}{\epsilon_s} - 1 \right)} \left(\bar{T}_w^4 - \bar{T}_s^4 \right)$$

$$+ I \Delta V \pi(r_0^2 - r_1^2) \Delta x. \quad (6)$$

Because the temperature variation across the tube wall was small in these experiments, no correction was included for this source of error in the data reductions. Calculated inside surface temperatures indicated that the inside temperature differed from the outside temperature by $\sim 0.2\%$ or less. No correction was included for the effect of axial expansion of the test section or for heat conduction from the test section to the radiation shield. Values of the numerically determined derivatives in Eq. (6) varied from 300 K/cm at the test-section inlet to -1350 K/cm at the test-section exit with the derivatives for the central portion of the test section being around 31 K/cm. The accuracy of these derivatives is best over the central por-

tion of the test section.

Properties for hydrogen and helium were taken from Refs. 8 and 9, respectively.

The measured inlet gas temperature was assumed to be the stream stagnation temperature, and the static inlet temperature was determined by trial and error using the equation

$$T_{f1} = T_{o1} - \frac{V^2}{2 \rho_{f1} A_1^2 C_{pf1}}. \quad (7)$$

This procedure was also used to obtain a static temperature at the exit. Knowing the inlet conditions and the flow rate together with the assumption that the axial pressure profile is linear makes it possible to determine the axial static enthalpy profile and thus the axial static bulk temperature profile by employing the expression

$$q_f = \left(h_{x+\Delta x} - h_x + \frac{V_{x+\Delta x}^2 - V_x^2}{2} \right) w. \quad (8)$$

In solving Eq. (8) for $h_{x+\Delta x}$, trial static fluid temperatures are used until agreement with the thermodynamic table value of $h_{x+\Delta x}$ for the static pressure and assumed static temperature at $x+\Delta x$ is achieved.

The assumption of a linear pressure-drop profile between the measured inlet and exit pressures makes the solution of Eq. (8) possible by trial-and-error method. For stagnation enthalpy Eq. (8) is directly solvable. To obtain values of the local heat-transfer coefficients, the heat-flux profile was interpolated to give the heat flux at each thermocouple location and the heat-transfer coefficients were then determined from

$$h_L = \frac{q_{fL}''}{(T_w - T_{fo})}, \quad (9)$$

where q_{fL}'' is the local heat flux, T_w is the measured wall temperature, and T_{fo} is the local stagnation temperature of the fluid. The static and stagnation temperatures of the coolant were essentially equal at the test-section inlet and differed by less than 3% at the test-section exit for all runs. Calculation also reveals that the adiabatic wall

temperature and the fluid stagnation temperature would not differ by more than 1% for the conditions of the experimental data so that an assumed recovery factor of unity, implied by Eq. (9), is quite reasonable. Only the local values of heat-transfer coefficient are discussed and presented in this paper, although the average value for each axial increment was also determined during data reduction.

THERMOCOUPLE CALIBRATION

The test-section wall thermocouples were calibrated by determining the change in electromotive-force (emf) output of each thermocouple when the power leads to the test section were reversed. To avoid damaging the test section at high power settings, the rapid switching technique of Davenport and Magee (10) was not used. Instead, a series of runs was made at a constant flow rate but with different power settings with one polarity, and the runs were then repeated with the polarity reversed. Very accurate flow, power, and exit-gas temperature indications were obtained with excellent repeatability by using the computerized data-monitoring system, and the method yielded reasonable calibration for all thermocouples except those with small errors of the order of ± 3 K. However, because the voltage pick-up of the capacitance-welded thermocouples is proportional to the test-section voltage drop over the length covered by the thermocouple,

$$\Delta \epsilon \propto \rho_m \frac{\Delta x}{A_c} I, \quad (10)$$

and because the resistivity of the molybdenum tube material is very nearly directly proportional to the temperature, the thermocouple error can be expressed as

$$\Delta \epsilon \propto \frac{\Delta x}{A_c} IT, \quad (11)$$

which indicates that a plot of thermocouple error against IT should be linear.

Typical plots of thermocouple error against IT where T was determined at the average emf output of the thermocouple are shown in Fig. 9. Clearly the thermocouples behaved as predicted. For data reduction a least-squares line was fit through the calibration data for each thermocouple.

Fig. 9. Typical thermocouple calibration curves.

RESULTS AND DISCUSSION

Experimental axial wall-temperature profiles for helium are shown in Fig. 10, and three representative profiles for hydrogen are shown in Fig. 11. Note that with increased power input to the test section at constant flow rate, the wall-temperature distributions change from a linear variation with distance to a higher-order nonlinear variation. The two lower curves in Figs. 10 and 11 correspond to cases wherein the heat flux to the test section was nearly constant with axial distance.

To compare the heat-transfer coefficients of this study with the results of previous investigators the heat-transfer-correlation equation proposed by M. J. Taylor (1) was used:

$$Nu_b = 0.023 Re_b^{0.8} Pr_b^{0.4} \left(\frac{T_w}{T_b} \right) \exp \left(-0.57 - \frac{1.59}{x/D} \right). \quad (12)$$

Fig. 10. Typical wall temperature profiles for helium.

Taylor has shown that beyond the immediate vicinity of the critical point of hydrogen, Eq. (12) agrees within $\pm 25\%$ for 87% of 3674 data points taken from various authors. In addition, for helium and nitrogen data, Eq. (12) predicted heat-transfer coefficients that agreed to better than $\pm 25\%$ for the 98% of the 447 points considered. The helium and hydrogen data obtained in this investigation are compared with Eq. (12) in Figs. 12 and 13, respectively. In these figures, the fluid properties were evaluated at the bulk static coolant temperature T_b . During these comparisons it was found that the heat-transfer coefficients for $x/D < 9$ did not correlate well with Eq. (12) and these points as well as those very near the test-section exit ($x/D > 74$) were excluded from Figs. 12 and 13. Figure 12 shows that Eq. (12) correlates the helium data of this investigation for $9 < x/D < 74$ to better than $\pm 10\%$ over the full range of the data. For the hydrogen data, Fig. 13, the correlation is no better than $\pm 15\%$ with most of the points falling below the correlation line at Reynolds numbers greater than 2×10^4 .

Three experimental heat-flux profiles for hydrogen shown in Fig. 14 were selected for comparison with numerical calculations. These comparisons

Fig. 11. Typical wall temperature profiles for hydrogen with Test Section 2.

should be viewed as preliminary, and the conclusions drawn from them should not be considered final. Much more exhaustive and systematic comparisons with carefully taken data over a wider range than used in this experimental investigation would be needed before definite conclusions as to the adequacy of the prediction capability of the numerical approach could be reached. A nearly constant heat-flux condition was found to exist for the low heat-flux case, whereas the heat flux varied by a factor of ~ 4.0 for the highest heat-flux case. The numerical code of C. A. Bankston and D. M. McEligot (6) was chosen for comparison with the experimental results. This code incorporates an implicit solution to the coupled two-dimensional boundary-layer equations for either laminar, turbulent, or mixed flows in circular tubes. The turbulence model incorporated in the code is that of Van Driest (see Ref. 6 for a complete discussion of turbulence models) because it was shown (by Refs. 5,6) to produce the best agreement with experimental data. In addition, the code uses Reynolds analogy to relate the turbulent diffusivities of heat momentum. Properties of hydrogen over the range of the experimental results were approximated by the power-law functions,

Fig. 12. Local heat-transfer data for helium at the thermocouple locations.

Fig. 13. Local heat transfer data for hydrogen at the thermocouple location $9 < x/D < 74$.

$$\frac{\rho}{\rho_1} = \frac{T_1}{T}$$

$$\frac{\mu}{\mu_1} = \left(\frac{T}{T_1} \right)^{0.7}$$

$$\frac{k}{k_1} = \left(\frac{T}{T_1} \right)^{0.7}$$

$$\frac{C_p}{C_{p1}} = 1.0$$

Over the temperature range of interest these functions represent the true properties very closely. Typical errors that can be expected from functions in the temperature range from 300 to 700 K are 3% in C_p , 1% in density, and 2% in viscosity and thermal conductivity.

A uniform velocity profile was assumed at the inlet of the test section which was 28 diameters upstream of the start of heating.

A comparison of predicted wall and bulk temperature profiles with experimental results (Fig. 15) shows that the numerical code underpredicts the experiment at all axial positions. Also, the agreement becomes worse as the heat flux to the tube increases and the discrepancy reaches a maximum of ~11% at the exit of the tube for the highest heat-

Fig. 14. Experimental heat-flux profiles for hydrogen used for comparison with numerical analysis.

flux case.

Figure 16 shows the predicted and experimental axial wall-temperature profiles normalized to the inlet bulk temperature for the three heat-flux cases. The agreement is ~5% for the low heat-flux case, ~10% for the intermediate case, and ~20% for the highest heat-flux case.

Similar comparisons made by Bankston and McEligot (6) for nitrogen runs of Perkins and Worsae-Schmidt (11) showed that the code just overpredicted the

Fig. 15. Comparison of axial wall-to-bulk stagnation temperature profiles for hydrogen for the three heat flux profiles of Fig. 14.

Fig. 16. Comparison of experimental and analytical axial wall-temperature profiles for hydrogen for the three heat fluxes of Fig. 14.

wall temperature near the inlet and underpredicted it further downstream.

The variations of Nusselt number with axial position in the test section are shown in Fig. 17. As indicated, the code overpredicts the Nusselt number for all heat-flux cases, at every axial position. Agreement is worst for the intermediate heat-flux case where the code predicts Nusselt numbers that are $\sim 20\%$ too high. From these comparisons it appears that the turbulence model used in the code, which was developed from adiabatic, fully developed flow measurements, may not be adequate for cases where

Fig. 17. Comparison of Nusselt numbers for hydrogen for the three heat-flux profiles of Fig. 14.

developing heated flows are present. Note, however, that the numerical results do predict the trend of the data in all cases, which indicates that the data may have a systematic error.

A final comparison of results is presented in Fig. 18, which shows the degree to which Taylor's correlation and the numerical code of Bankston and McEligot predict the variation of Nusselt number with wall-to-bulk temperature ratio. From Fig. 18 it is clear that the exponent on (T_w/T_b) in Taylor's correlation is larger than that which would agree well with our data and that the numerical code predicts a value that is less. For comparison, a line corresponding to $(T_w/T_b)^{-0.3}$ is included in Fig. 18. An exponent of -0.3 on the wall-to-bulk-temperature

Fig. 18. Comparison of experimental results at $x/D = 73.4$ to numerical calculations and Taylor's correlation.

ratio is consistent with the results of McCarthy and Wolf (12) for low values of wall-to-bulk temperature ratio (1.5 to 2.8).

SUMMARY

Results of an experimental investigation of heat transfer to hydrogen and helium at moderate wall temperatures and heat fluxes are presented.

These results are compared with the most recent correlation of similar data by M. F. Taylor, and it is shown that the data agree well with this correlation. Minor changes in the constants of Taylor's correlation would improve the agreement.

Comparison of selected runs for hydrogen with a two-dimensional numerical code developed by Bankston and McEligot show that the code does predict the data trends but is consistently low in predicting wall temperatures and consistently high in predicting Nusselt numbers.

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