HEAT TRANSFER TO AVIATION FUELS

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الانتقال الحراري لوقود الطيران

تم قياس معاملات الانتقال الحراري لانواع من الوقود المستعمل في الطائرات النفاثة باستخدام مبدل حراري ذي أنبوبة واحدة في حالات تمثل تلك التي يتعرض لها الوقود أثناء الطيران ، وكذلك تم اقتراح معادلة يمكن بها حساب هذه المعاملات وقد تبين أن قيمة معامل الانتقال الحراري للوقود ذي الوقاية العالية ضد الاشتعال عند سقوط الطائرات هي أعلى منها للانواع الاخرى من الوقود .

Abstract

Heat transfer coefficients of conventional and FM-4 safety aviation fuels were measured when passed through a single tube heat exchanger. This reproduces the conditions to which aviation fuel will be subjected in the fuel-cooled oil coolers of the supersonic transport. It was found that at near ambient temperatures the sheared FM-4 safety aviation fuel and the conventional fuel have very similar heat transfer properties over a Reynolds number range 200-10,000.

The heat transfer results for the turbulent region were compared with the empirical relationship of Sieder and Tate and the experimental data were found to vary somewhat from the calculated values. A modified Sieder and Tate equation was found to correlate the data better. The heat transfer coefficient of the modified safety fuels was found to be remarkably higher than the other two fuels at the same flow and temperature conditions.

Introduction

In a low speed aircraft crash, the fuel escaping from a ruptured fuel tank is transformed into a highly inflammable fuel mist, which is responsible for the high incidents of fires following crashes. By the addition of 0.3% of a high molecular weight polymer (e.g. FM-4 produced by Shell Co.) to conventional aviation turbine (Avtur) fuel, the required non-misting property is obtained.

In supersonic aircraft, the fuel is used as a coolant fluid in the engine system and airframe components, flowing at Reynolds numbers ranging from 1500 at let-down to 12,000 at take-off. Recent filtration tests have indicated that it will be necessary to shear the modified fuel to a great extent in order to pass it through the engine fuel filters. Consequently; the heat transfer properties have been evaluated on samples of unmodified Avtur, FM-4 polymer fuel and on a sample of highly sheared FM-4 fuel, in order to know the effect of the addition of the polymer and shearing on the heat transfer characteristics of the fuels.

EXPERIMENTAL PROCEDURE

Factors Affecting the Design of the Apparatus

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The main requirement of the test rig was that it should measure the heat transfer coefficient as accurately as possible, without highly shearing the fuel and causing degradation of the polymer. Thus, a gas drive system using helium was employed instead of the conventional fuel pump, followed by separating the gas from the fuel in a bladdered fuel reservoir.

The single heat transfer tube was heated electrically, since this offers much better temperature control and measurement than would a hot oil bath system. A vertical tube assembly was used, which offers better temperature control in the region of Re = 2200 [1].

A digital voltmeter was used instead of the conventional potentiometer, so that rapid and easy temperature observations could be made, in order to establish the required fuel temperature rise by adjustment of the electrical power input.

General Description of the Rig

A helium cylinder fitted with a reducing valve pressurises the bladder in a modified 'Greer Mercier Hydraulic Accumulator' The fuel surrounding the bladder is forced out by expansion of the bladder. The helium supply line is fitted with two valves, one to the atmosphere, and one to a vacuum pump. These enable the bladder to be depressurised and subsequently collapsed, so that the fuel reservoir can be refilled, as shown in Fig. 1.

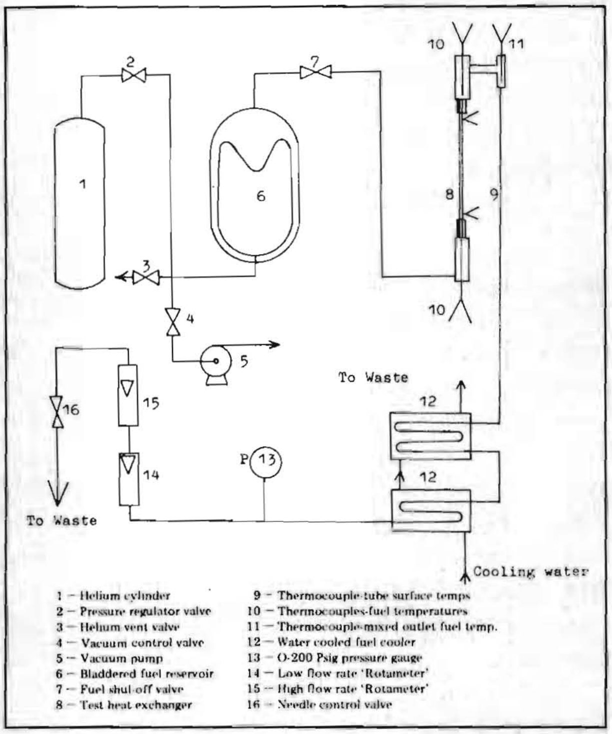

FIG. 1 Schematic Diagram, "Single Tube Heat Transfer Rig."

The fuel is forced from the reservoir through the single tube heat exchanger and then through two water coolers in series. These are simply stainless steel pipes, surrounded by a copper jacket through which cold water is passed. Constant effluent fuel temperature was required since the calibration of the two variable area flowmeters, downstream from the coolers, was dependent on the fuel temperature. To cover the wide range of flowrates of all three fuels used, two flowmeters were employed, one measures low flow rates and the other measures high flow rates with the necessary corrections for the different fuels used. The fuel flow rate is controlled by a needle valve from which effluent flows to a collector drum. Pressure gauges on the pressure regulator and after the heat transfer tube show the gas drive pressure and the fuel pressure in the rig. The rig was tested to a maximum pressure of 150 psig. This being necessary for the most viscous fuel, at high flow rates. All fuel pipelines were made of 1/4" O.D. stainless steel tubing joined with 'Ermeto' stainless steel couplings. The use of certain metals were restricted due to their catalysis of fuel degradation reactions [2].

Heat Exchangers

Figure 2 illustrates the heat exchanger assembly in use. The heat exchangers employed were single vertically mounted stainless steel tubes of two sizes, 0.092" ID and 0.047" ID. In order to measure tube wall temperatures, 0.0124 in. diam. constantan thermocouples were silver-soldered to the outer tube surface, 1/4" from each end of

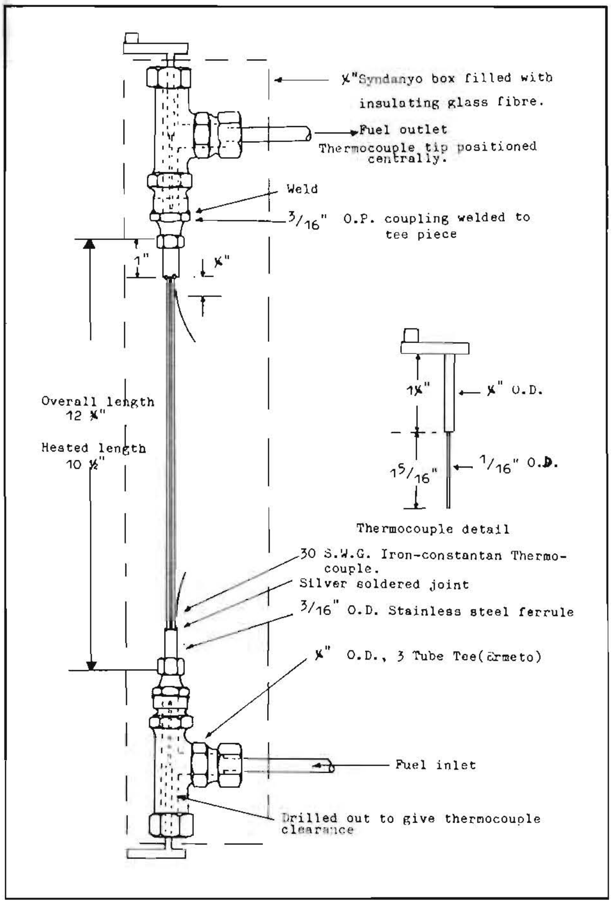

FIG. 2 Single tube heat exchanger.

the tube. Heat was generated by passing a low voltage alternating current through the tube, and was dissipated by the fuel flowing through the tube and by heat losses to the outside. Iron-constantan thermocouples, encased in 1/16" O.D. 1 $\frac{5}{16}$" long stainless steel sheaths, used to measure fuel inlet and outlet temperatures, were positioned centrally, and in close proximity to the inlet and outlet of the heat exchanger tube. The whole heat exchanger assembly was held in an insulating box made of 1/4" hard asbestos (Syndanyo), by 1/4" bolts, welded to the couplings. This box was packed with glass fibre insulation to minimise heat losses from the tube.

Temperature Measurement

The themocouples in the heat exchanger assembly were connected to a selector switch, so that, the voltage generated by the thermocouples could be observed on a solartron model LM1420.2 digital voltmeter, which enabled the voltage to be read to $\pm$ 0.01 mv. The use of the outside wall temperature, as measured, in evaluating the values of 'h' would not be strictly correct, since a temperature gradient exists across the tube wall. It is possible, however, to calculate the difference between the inside and outside wall surface temperatures in terms of heat flux, by making the following assumptions.

i) Constant electrical resistivity and thermal conductivity of tube material.
ii) Radial potential gradient within tube wall is negligible.
iii) Longitudinal temperature gradient is negligible.

The rate of heat generation throughout a cylinder of volume V is equal to: $\frac{q}{V}$

At steady state, assuming the flow of heat to be purely radial,

$$\frac{1}{r}\frac{d}{dr}\left(r\frac{dt}{dr}\right) + \frac{q}{k.V} = 0$$

the general solution of which is [3]:

$$t = A + B\log_e r - 1/4\, r^2 \frac{q}{k.V}$$

For the cylindrical tube under consideration (Figure 3), with internal and external radii r_2 and r_1, and internal and external temperatures of t_2 and t_1, the general solution becomes:

$$(t_2 - t_1) = B\log_e \frac{r_2}{r_1} - \frac{(r_2^2 - r_1^2)}{4\,k.\,V} \qquad (1)$$

The constant B can be determined from the boundary condition at the inner surface $r = r_o$ as there is no loss of heat at the imaginary adiabatic surface, thus

$$\left(\frac{\partial t}{\partial r}\right)_{r = r_o(\text{inner})} = \frac{B}{r_o} - \frac{r_o\, q}{2\,k.V} = 0$$

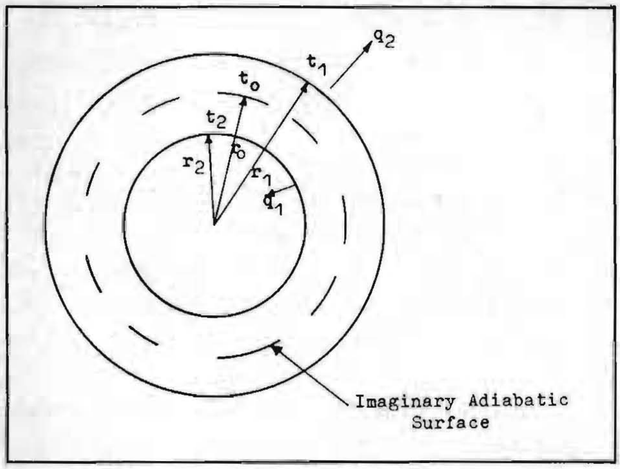

FIG. 3 Section Through Heat Transfer Tube.

Equation (1) then becomes:

$$k.(t_0-t_1) = \frac{q_2}{2\pi L}\left(1/2 - \frac{r_0^2}{r_1^2-r_0^2}\log_e \frac{r_1}{r_0}\right) \qquad (2)$$

Also, from the boundary condition at the outer surface $r = r_{0(outer)}$, equation (1) becomes:

$$k.(t_2-t_0) = \frac{q_2}{2\pi L}\left(1/2 - \frac{r_0^2}{r_0^2-r_2^2}\log_e \frac{r_0}{r_2}\right) \qquad (3)$$

At constant potential v,

$$\text{Heat dissipated} = \frac{v^2 . A}{p \cdot L} \text{ per second}$$

Therefore the rate at which heat is generated is proportional to A when v, p and L are fixed.

Assuming zero potential gradient within the wall, all the heat lost from the outside surface of the tube (q_2) must be generated within the cylinderical section bounded by the outside surface and the imaginary adiabatic surface. Similarly the heat lost to the fuel flowing through the tube (q_1) must be generated within the cylinderical section bounded by the inside surface of the tube and the imaginary adiabatic surface. Therefore

$$\frac{q_1}{q_2} = \frac{r_0^2 - r_2^2}{r_1^2 - r_0^2} \qquad (4)$$

It follows therefore, that for any measured outside wall temperature, the temperature of the inside wall may be calculated from equations 2,3 and 4.Putting:

$$R : \frac{\text{heat to fuel}}{\text{heat loss}} = \frac{q_1}{q_2} \qquad (5)$$

rearranging equation 4, to find r_0 as a function of R, r_1,r_2.

$$r_0^2 = \frac{R\,r_1^2 + r_2^2}{1 + R}$$

Eliminating t_0 from equations (2) and (3), thus

$$t_2-t_1 = \frac{v\,I}{4\pi k L}\left(1 - \frac{2(Rr_1^2 + r_2^2)}{(1+R)(r_1^2-r_2^2)} \ln \frac{r_1}{r_2}\right) \qquad (6)$$

Hence, by determining the rate of flow of heat to the fuel, and the rate of heat loss from the tube, the temperature difference across the tube can be deduced. The magnitude of the temperature corrections obtained by this method were insignificant for all heat fluxes below 45 Chu hr^{-1} and ranged to values as large as 1.6°C at a heat flux of 820 Chu hr^{-1}.

Electrical System:

A 2 K.v.A. variac was powered from a 230 volt A.C. single phase mains supply, to provide a variable A.C. voltage (0-250 volts) onto which was superimposed fine control by a 0.5 K.v.A. variac operating through a 10 : 1 step down transformer, with a maximum secondary output of 200 ampere at 10 volts, as shown in Fig. (4).

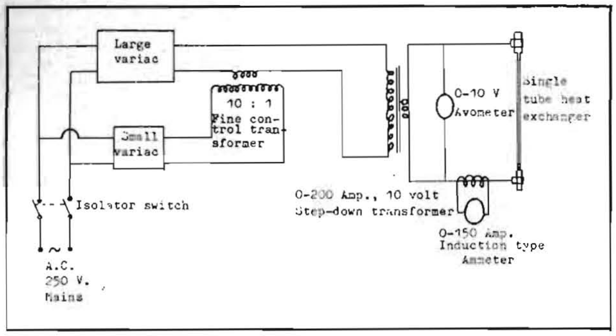

FIG. 4 Electrical power system.

Test Fuels:

Two samples of fuel were used:

SPL369/72	Unmodified Avtur
SPL368/72	FM-4 Modified Avtur

The third test fuel was derived by shearing the modified fuel, by passing it once through a pump diesel injector system. This fuel is subsequently referred to as 'the sheared fuel'.

Test and Methods:

In order to carry out experiments in both the streamline and turbulent flow conditions, the two heat transfer tubes of I.D. 0.092" and 0.047" were used to raise the temperature of the fuel by 5°, 10°, 15° and 20°C where possible at each of the flow-rates of 5, 10, 20, 40, 60, 80 lb. hr^{-1}.

After the use of one fuel, the apparatus was flushed with the next fuel prior to use. The order of usage of fuels was unmodified fuel, sheared modified fuel, unsheared modified fuel so that the polymer, which adheres to the inside of the fuel reservoir and lines did not comtaminate the unmodified fuel. When it was necessary to perform experiments on the unmodified fuel after the modified fuel, the fuel reservoir was dismantled and thoroughly cleaned. After flushing the rig with clean unmodified fuel, the experiments could then be performed without risk of polymer contaminations.

Results and discussion

The full range of experiments have been carried out on the unmodified Avtur and the sheared FM-4/Avtur at flow rates of 5, 10, 20, 40, 60, and 80 lbs/hr. with temperature differences of 5, 10, 15, 20°C in both of the heat transfer tubes. From the figures obtained, the respective heat transfer coefficients have been calculated as

$$h = \frac{q}{A\ \Delta tm} \tag{7}$$

where A is the heat transfer area, the inside tube area and

$$\Delta tm = \frac{(t_4 - t_3) - (t_2 - t_1)}{\log_e \dfrac{t_4 - t_3}{t_2 - t_1}} \tag{8}$$

where: t_1 = fuel inlet temperature

t_2 = calculated inside tube wall temperature at inlet end.

t_3 = fuel outlet temperature.

t_4 = calculated inside tube wall temperature at outlet end.

The values obtained show that at similar flow rates the heat transfer coefficient for the unmodified fuel is approximately 20% larger than that for the sheared FM-4/fuel (Table 1).

For further calculations the viscosity of the fuel is required. However, the concept of viscosity for both the modified fuel samples is rather doubtful decreasing with increasing rate of shear. Hence, the values obtained from the low shear rate viscosity measurements made at three different temperatures do not necessarily relate to the heat transfer experimental conditions. This effect is clearly shown to exist when the viscosities are plotted on an A.S.T.M. standard viscosity-temperature chart. Using the interpolations obtained

TABLE 1 Comparison of "h" values for modified, unmodified and sheared fuels

0.047" I.D. Tube		Unmodified fuel			Sheared fuel			Modified fuel		
Nominal conditions		Mass flowrate lb/hr	"h" Chu/ft^2 hr	Re.	Mass flowrate lb/hr	"h" Chu/ft^2 hr°c	Re.	Mass flowrate lb/hr	"h" Chu/ft^2 hr	Re.
5 lb/hr approx.	5°C	4.85	209	600	5.1	158	538	4.6	231	197
	10°C	4.65	218	593	4.9	187	569	4.4	239	203
	15°C	4.65	223	641	4.82	196	605	4.65	259	227
	20°C	4.85	241	723	4.78	213	653	4.4	235	228
10 lb/hr	5°C	9.7	333	1134	10	277	1029	—	—	—
	10°C	9.8	354	1211	10	303	1141	8.9	441	384
	15°C	9.9	351	1247	10	312	1246	10.1	305	480
	20°C	9.9	402	1585	10	316	1366	10.6	442	465
20 lb/hr	5°C	20.3	563	2374	19.8	373	2050	—	—	—
	10°C	20.4	606	2447	20	470	2273	21.0	584	896
	15°C	20.6	666	2838	20.2	521	2478	19.7	517	892
	20°C	20.4	778	3042	20.2	569	2714	—	—	—
40 lb/hr	5°C	38.4	1504	4489	39.3	1108	4043	—	—	—
	10°C	39.1	1514	4985	37.3	1131	4436	42.3	618	1797
	15°C	38.4	1602	5290	37.3	1307	4576	40.4	612	1767
	20°C	39.1	1691	6111	37.9	1358	5049	39.2	571	1862
60 lb/hr	5°C	61.5	2191	7841	58.6	1840	6029	—	—	—
	10°C	60.8	2178	8002	58.9	1810	6558	63.8	1010	2689
	15°C	61	2365	8660	58.3	1900	7152	62.1	971	2694
	20°C	60.3	2495	9425	56.3	1980	7532	60.0	890	2715
80 lb/hr	5°C	77.6	2482	9309	84.5	2340	8835	—	—	—
	10°C	75.3	2841	10370	—	—	—	—	—	—
	15°C	74.3	2838	10550	—	—	—	78.6	1161	3410
	20°C	—	—	—	81	3040	10970	—	—	—

Table 1 continued

0.092" I.D. Tube		Unmodified fuel			Sheared fuel			Modified fuel		
Nominal conditions		Mass flowrate lb/hr	"h" Chu/ft^2 hr	Re.	Mass flowrate lb/hr	"h" Chu/ft^{2}hr°C	Re.	Mass flowrate lb/hr	"h" Chu/ft^{2}hr	Re.
5 lb/hr	5°C	5.0	140	282	4.5	135	219	5.25	52.2	115
	10°C	5.0	139	303	4.5	131	228	5.25	55.0	117
	15°C	5.0	144	336	4.5	131	251	5.25	60.1	122
	20°C	5.0	145	322	4.5	135	237	5.25	57.0	123
10 lb/hr	5°C	10.0	199	583	10.6	206	513	10.5	118.0	114
	10°C	10.0	198	606	10.7	200	539	10.5	118.0	115
	15°C	10.0	188	674	10.8	199	584	10.5	121.0	119
	20°C	10.0	193	632	10.9	202	601	10.7	129.0	123
20 lb/hr	5°C	20.0	108	1190	19.8	126	903			
	10°C	20.0	109	1210	19.8	140	1054			
	15°C	20.0	92.4	1285	19.4	174	1089			
	20°C	—	—	—	—	—	—			
40 lb/hr	5°C	40.0	246	2356	40.2	115	1896			
	10°C	40.0	259	2425	39.9	226	2045			
	15°C	40.0	261	2526	39.3	263	2124			
	20°C	—	—	—	39.3	255	2154			
60 lb/hr	5°C	59.8	429	3485	59.8	378	2790			
	10°C	59.8	460	3665	59.5	389	3069			
	15°C	59.6	462	3764	60	420	3256			
	20°C	59.8	498	4074	59.8	429	3425			
80 lb/hr	5°C	—	—	—	80	500	3943			
	10°C	79.2	595	4800	80	544	4127			
	15°C	—	—	—	78.7	575	4416			
	20°C	—	—	—	78.1	598	4633			

from this chart the Reynolds number* was calculated for each experiment. The range of Reynolds number covered was from 300 to 10500. For Newtonian fluids a plot of heat transfer coefficient against Reynolds number is a soundly based comparison, since Reynolds number is an indicating factor of the flow conditions existing in the tube. The results (Figs. 5 and 6) show the expected curve, with a greater scatter of the data at a Reynolds number about 3,500. A comparison based on Reynolds number show that the sheared modified fuel has slightly lower heat transfer coefficients than the unmodified fuel at the same Reynolds number.

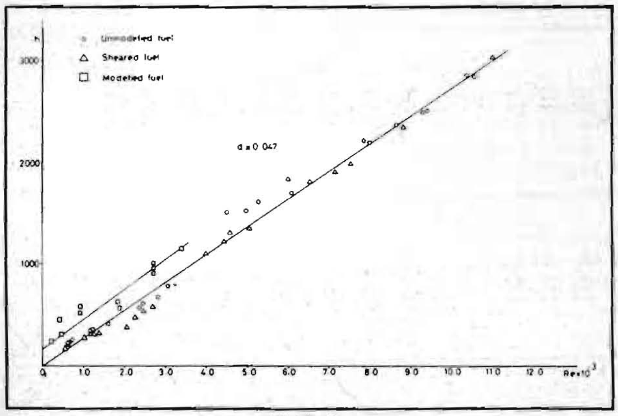

FIG. 5

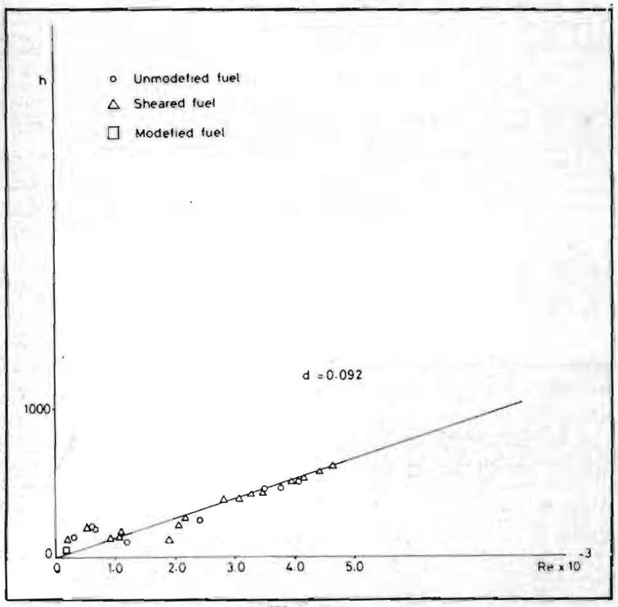

FIG. 6

The values obtained by experiment in the turbulent region were then compared with those predicted by general correlations for heat transfer inside heated pipes, equation due to Sieder and Tate [4]. i.e.

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.33} \left(\frac{\mu b}{\mu w}\right)^{0.14} \qquad (11)$$

and a plot of Nu versus $Re^{0.8}\, Pr^{0.33}$

$$\left(\frac{\mu b}{\mu w}\right)^{0.14}$$

gave a straight line, not passing through the origin, with a larger slope than expected, for both the unmodified and the sheared fuel, (Fig. 7).

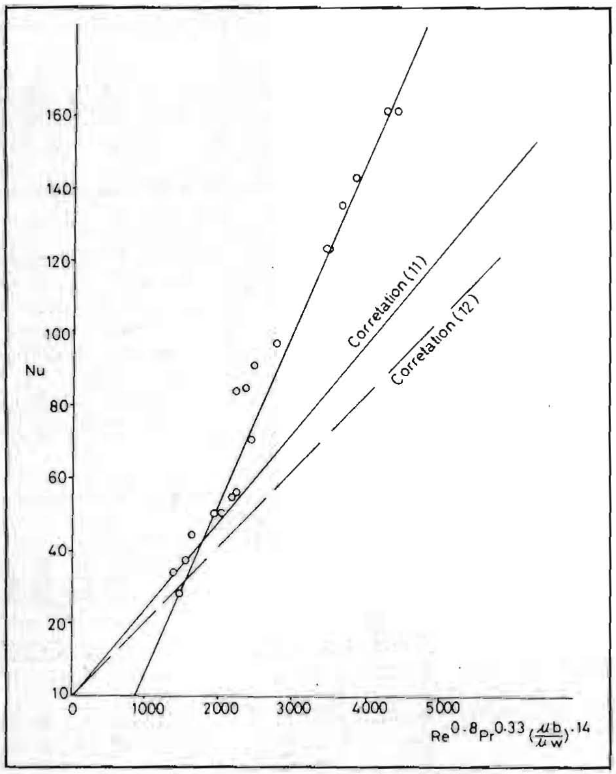

FIG. 7 Comparion of Experimental Results for Unmodified and sheared Fuel.

A proposed correlation for Avtur at higher temperatures [1],

$$Nu = 0.0207\, Re^{0.925} \qquad (12)$$

was found to be in similar disagreement with the experimental results, (Fig. 7). Since the variation in $Pr^{0.33}$ and $\left(\frac{\mu b}{\mu w}\right)^{0.14}$ were

* *For the calculation of Reynolds number, values of the kinematic viscosity correspond to the average temperature between inlet and outlet.*

small, it was assumed that the indices of these terms were applicable to these results. Hence a graph of log $Nu/Pr^{0.33}\ (\frac{\mu b}{\mu w})^{0.14}$ versus log Re yielded the experimental value of 1.14 for the Reynolds Number Index so that a suggested correlation for these results is:

$$Nu = 0.00165\ Re^{1.14}\ Pr^{0.33}(\frac{\mu b}{\mu w})^{0.14} \quad (13)$$

This applies for both the sheared fuel and the unmodified fuel (Figure 8).

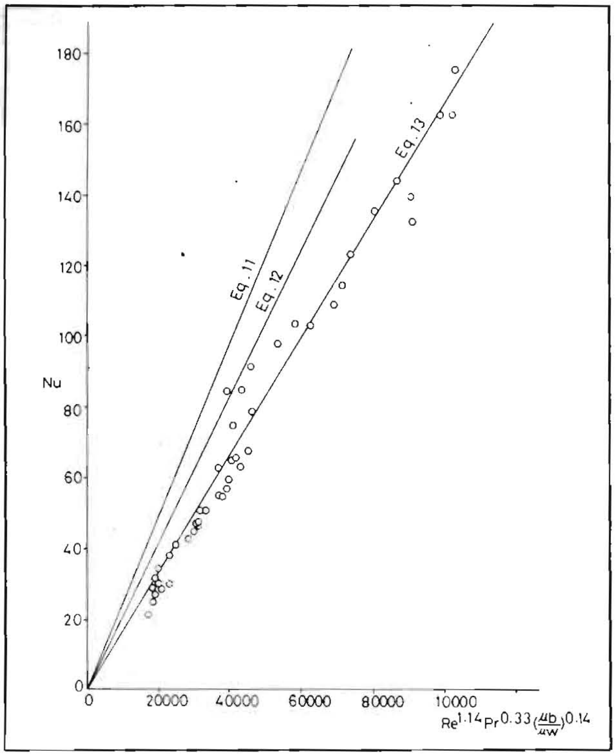

FIG. 8 Plot for approximate correlation $Nu \propto Re^{1.14}Pr^{0.33}(\frac{\mu b}{\mu w})^{0.14}$ Showing constant of proportionality to be 0.00165.

It was thought that the difference in the coefficients may possibly have been due to the fact that the larger tube has a quite small L/D ratio, hence entrance effects would be present. However, this is not true since the smaller heat transfer tube, with L/D = 255, gives results which correlate with the same Reynolds Number Index. The Sieder and Tate equation generally applies for fully developed turbulent flow [5] and the results obtained here are in the critical and turbulent regions. Hence this may be a possible reason for the variation in indices. This does not however account for the variation in the constant term, the reason for which is unknown.

Experiments on the modified fuel were found to be inconsistent in two respects. The fuel outlet temperature was observed to oscillate by up to 6°C. A possible cause was the existence of inhomogenities in the fuel due to a build-up of polymer on the inside tube wall surface. Evidence for this phenomenon was obtained by observing fuel, impinging directly from a capillary tube onto a glass plate, under strong light. The fuel was observed to be more viscous in the outer regions of the resulting flow down the glass plate.

This temperature oscillation may however have been partially due to that the fuel outlet thermocouple did not indicate a true mixed fuel temperature. This is believed to have been due to the viscoelastic properties of the fuel, which leads to streamline flow even at very high mass flowrates. The thermocouple tip, slightly downstream from the tube outlet was measuring the temperature of a certain portion of the outlet stream with slightly different thermocouple positionings, exceptionally high and exceptionally low fuel outlet temperatures were recorded. This idea was confirmed by the positioning of an additional thermocouple downstream of the fuel outlet, where full mixing had occurred. However, at higher apparent Reynolds Number the temperature fluctuations diminished so that the results obtained from the smaller diameter tube are more reliable.

The results (Table 1) indicate that heat transfer to the modified fuel exceeds that for the sheared modified fuel and Avtur at a given Reynolds number (estimated from the measured low shear viscosity). However, since in aircraft, the flow rate is fixed by engine fuel demand or operating speed, a more meaningful comparison is shown in Table 1 for values of 'h' at different mass flow rate. This indicates a larger adverse effect on heat transfer the more the fuel deviates from Newtonian behaviour.

NOMENCLATURE

Symbol	Name	Units
h	Heat transfer coefficient	Chu/hr ft^2°C
I	Current	amp
k	Thermal conductivity	Chu/hr ft^{-1}°C
L	Length	ft
q	Heat flow rate	Chu/hr
r	Radius	ft
R	Temperature correction factor	—
t	Temperature	°C
tmean	Mean fuel temperature	°C
v	Voltage	volt
V	Volume	—
Δt	Temperature difference	°C
Δtm	Logarithmic temperature difference	°C
$^{\pi}b$	Bulk absolute viscosity	lb/ft hr.
$^{\pi}w$	Wall absolute viscosity	lb/ft hr.
Nu	Nusselt number	—
Pr	Prandtl number	—
Re	Reynolds number	—
p	density	

Subscripts: b evaluated at bulk mean temperature
w evaluated at mean wall temperature.

References

1. *Smith, J.D. — I. & E.C. Proc Des. & Dev. 8, 299, (1969).*

2. *Smith, J.D. — Aircraft Engineering, 39, No. 4, 19 (1967).*

3. *Mc Adams, W.H. — "Heat Transmission", McGraw Hill, New York, 1954.*

4. *Sieder, E.N. & Tate, C.E. — Ind. Eng. Chem. 18, 1429, (1936).*

5. *Krieth, F. — "Principles of Heat Transfer, 2nd Edition, International Textbook Company, London, (1972).*