THERMAL PERFORMANCE TESTING OF HEAT EXCHANGERS
WITH REFERENCE TO FUTURE AUTOMATION

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SUMMARY

The project was sponsored by an industrial company and was an investigation into reducing the time spent in thermal performance testing heat exchangers. Conventional testing requires establishing steady state conditions which is time consuming. The research considered two approaches - use of a non-steady state testing technique and closed-loop control of steady state testing.

After reviewing and assessing existing non-steady state testing techniques the work starts to investigate a new dynamic testing method. In this method the objective was to develop a model of a heat exchanger from measurement of the inlet and outlet temperature responses following a temperature input disturbance. Then, to use the model in conjunction with only part of the transient response to predict the final steady state temperatures, thereby reducing testing time. Unfortunately, the time available limited the analysis to the first few steps in the development of the model and therefore the feasibility of dynamic testing has not been fully evaluated.

The second aspect of the research is contained in an appendix and gives a preliminary assessment of automation of the steady state testing of heat exchangers.

The report recommends that the sponsoring company should note the inconclusive investigation into one form of dynamic testing and consider closed-loop control as an alternative to reduce testing time.

KEY WORDS: HEAT EXCHANGERS, TESTING, REVIEW, CONTROL.

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NOMENCLATURE

A	HEAT TRANSFER SURFACE AREA	m^2
A_f	FLOW AREA	m^2
A_i	TOTAL SURFACE AREA INSIDE TUBES	m^2
A_o	TOTAL SURFACE AREA OUTSIDE TUBES	m^2
A_s	CROSS SECTIONAL AREA OF TEST CORE SOLID	m^2
c_f, c	SPECIFIC HEAT CAPACITY OF FLUID AT CONSTANT PRESSURE	kJ/kgK
c_s	SPECIFIC HEAT CAPACITY OF SOLID	kJ/kgK
D	HYDRAULIC DIAMETER	m
d_i	TUBE INSIDE DIAMETER	m
d_o	TUBE OUTSIDE DIAMETER	m
h	PARTIAL HEAT TRANSFER COEFFICIENT	kW/m^2K
h_i	INSIDE (OR TUBESIDE) PARTIAL HEAT TRANSFER COEFFICIENT	kW/m^2K
h_o	OUTSIDE (OR SHELLSIDE) PARTIAL HEAT TRANSFER COEFFICIENT	kW/m^2K
h_x	LOCAL HEAT TRANSFER COEFFICIENT	kW/m^2K
k	THERMAL CONDUCTIVITY	kW/mK
k_s	THERMAL CONDUCTIVITY OF TEST CORE SOLID	kW/mK
L	LENGTH OF TEST SECTION OR TUBE LENGTH	m
$\dot{m}$	MASS FLOW RATE	kg/s
$\dot{Q}$	HEAT TRANSFER (OR HEAT DISSIPATION) RATE	kW
r_{if}	INSIDE TUBE FOULING RESISTANCE	m^2K/kW
r_{of}	OUTSIDE TUBE FOULING RESISTANCE	m^2K/kW
Δr	TUBE WALL THICKNESS	m
t	TIME	s
U	OVERALL HEAT TRANSFER COEFFICIENT	kW/m^2K

v	VELOCITY	m/s
W_f	MASS OF FLUID CONTAINED IN TEST SECTION	kg
W_s	MASS OF TEST CORE SOLID OR MASS OF HEAT CAPACITOR CYLINDER	kg
x	AXIAL DISTANCE CO-ORDINATE	m

GREEK SYMBOLS

θ	TEMPERATURE	^oC
θ_a	AMBIENT TEMPERATURE	^oC
θ_b	MEAN BULK FLUID TEMPERATURE	^oC
θ_{c1}	COLD FLUID INLET TEMPERATURE	^oC
θ_{c2}	COLD FLUID OUTLET TEMPERATURE	^oC
θ_f	FLUID TEMPERATURE OR BOUNDARY LAYER FILM TEMPERATURE	^oC
θ_{f1}	INLET FLUID TEMPERATURE AT TIME = 0	^oC
θ_{f2}	OUTLET FLUID TEMPERATURE	^oC
θ_{h1}	HOT FLUID INLET TEMPERATURE	^oC
θ_{h2}	HOT FLUID OUTLET TEMPERATURE	^oC
θ_i	INITIAL INLET FLUID TEMPERATURE	^oC
θ_m	MEAN TEMPERATURE ABOUT WHICH THE TEMPERATURE OSCILLATES IN THE CYCLIC METHOD	^oC
θ_s	TEMPERATURE OF TEST CORE SOLID	^oC
θ_{si}	INITIAL TEMPERATURE OF HEAT CAPACITOR CYLINDER (EQUATION 1.1.)	^oC
θ_w	TUBE WALL TEMPERATURE	^oC
θ_1, θ_2	INLET AND OUTLET FLUID TEMPERATURES ON THE SAME SIDE OF THE HEAT EXCHANGER	^oC

$\Delta\theta_c$	$\theta_{c2} - \theta_{c1}$	K
$\Delta\theta_h$	$\theta_{h1} - \theta_{h2}$	K
$\Delta\theta_{LMTD}$	LOGARITHMIC MEAN TEMPERATURE DIFFERENCE	K
$\Delta\theta_m$	AMPLITUDE OF THE SINUSOIDAL TEMPERATURE VARIATION ABOUT θ_m	K
$\Delta\theta_1$	DEFINED IN FIG. I.1.	K
$\Delta\theta_2$	DEFINED IN FIG. I.2.	K
μ	DYNAMIC VISCOSITY	Ns/m^2
ρ	DENSITY	kg/m^3
ρ_f	DENSITY OF FLUID	kg/m^3
ρ_s	DENSITY OF TEST CORE SOLID	kg/m^3
ϕ	PHASE ANGLE	o
ω	FREQUENCY OF TEMPERATURE OSCILLATION	rad/s

DIMENSIONLESS PARAMETERS

jh	$= \frac{Nu}{Re.Pr^{\frac{1}{3}}} = St.Pr^{\frac{2}{3}}$	COLBURN j-FACTOR
L/D		LENGTH TO DIAMETER RATIO
NTU	$= \frac{hA}{\dot{m}_f c_f}$	NUMBER OF HEAT TRANSFER UNITS
Nu	$= \frac{hD}{k}$	NUSSELT NUMBER
Pr	$= \frac{c\mu}{k}$	PRANDTL NUMBER
R	= TEMPERATURE AMPLITUDE RATIO	
Re	$= \frac{\dot{m}D}{A_f\mu}$	REYNOLDS NUMBER
St	$= \frac{Nu}{Re.Pr}$	STANTON NUMBER
t*	$= \frac{hA}{W_s c_s}$	DIMENSIONLESS TIME
z	$= NTU.\frac{x}{L}$	DIMENSIONLESS DISTANCE

θ^*	$= \dfrac{\theta - \theta_1}{\theta_{f1} - \theta_{fi}}$	DIMENSIONLESS TEMPERATURE IN THE SINGLE-BLOW TECHNIQUE
θ'	$= \dfrac{\theta - \theta_m}{\Delta\theta_m}$	DIMENSIONLESS TEMPERATURE IN THE CYCLIC METHOD
λ	$= \dfrac{k_s A_s}{m_f c_f L}$	LONGITUDINAL CONDUCTION PARAMETER

ADDITIONAL SUBSCRIPTS

c	FIRST OR COLD FLUID STREAM
h	SECOND OR HOT FLUID STREAM
w	TUBE WALL
1,2,3	TO IDENTIFY DIFFERENT VARIABLES OF THE SAME TYPE

INTRODUCTION

In August 1976 the author embarked on a research project sponsored by Serck Heat Transfer and in conjunction with the Interdisciplinary Higher Degrees Scheme (I.H.D.) at the University of Aston in Birmingham. This introduction starts with a pre-amble about I.H.D. and Serck Heat Transfer. It then proceeds to outline the project, how it was approached and how the work is presented in this report. Finally, the introduction closes with a short background to the heat transfer concepts relating to Serck and the project.

I.1. PREAMBLE

I.1.1. THE I.H.D. SCHEME

The I.H.D. Scheme was set up in 1968 as a new kind of post-graduate training scheme linked to industry and the public service. The scheme gives the graduate the opportunity to tackle a real industrial problem using a multi-disciplinary approach, and, at the same time, providing a route to a Ph.D or M.Phil.

Within the scheme there are two streams. Firstly, the 'straight' I.H.D. stream, where the projects tackled usually involve the disciplines of Management, Economics or Social Science and give general experience in problem solving in business and technical areas. And, secondly, the total Technology (T.T.) stream, where the projects are of a technological nature and are undertaken by young engineers intending to make a career in professional engineering. The T.T. stream includes course work specifically on subjects influencing the work of the professional engineer.

The industrial experience gained during the course of the project is an important part of the scheme. Between 30%

and 70% of the students time is spent in the sponsoring organisation, with the remainder at the university.

I.1.2. THE SPONSORING ORGANISATION - SERCK HEAT TRANSFER

Serck Heat Transfer (S.H.T.) is one of eight U.K. operating companies in Serck Limited. In 1979, sales for the Serck Group totalled £91.9 million with S.H.T. contributing £21.4 million. The main S.H.T. factory is in Birmingham, employing about 1150 people, with others in Manchester and Hamburg.

The business activity of S.H.T. is heat exchange equipment. On the Birmingham site the company is divided into three manufacturing divisions, the Heavy Engineering Division (H.E.D.), the Engine Equipment Division (E.E.D.) and the Energy Systems Division. H.E.D.'s product range includes shell and tube heat exchangers and air cooled radiator type units for the marine, power generation, petro-chemical and large diesel engine industries. E.E.D. supplies small tubular oil and water coolers for the high speed diesel engine markets - trucks, tractors, earthmoving equipment - and oil coolers and fuel heaters to the worlds leading aero-engine manufacturers. The Energy Systems Division was formed in 1980 to exploit the opportunities in the field of industrial waste heat recovery and energy conservation.

Short term product and process development is undertaken within the operating divisions. Long term development and new product investigations are carried out as a central engineering function in the Advanced Engineering Department.

The research work was carried out in the Advanced Engineering Department, the company's Engineering Laboratory, and in the Department of Electrical Engineering at the University of Aston.

I.2. BACKGROUND TO THE PROJECT

Thermal performance testing of heat exchangers can form a significant part of the total work load for the company's Engineering Laboratory. At present, all the heat transfer test rigs are manually controlled with the requirement that steady state conditions must be established before measurement of the test variables. This results in a considerable amount of time being spent in testing with associated running costs being incurred. Therefore, there is a need to reduce testing time. In section A6.2 of Appendix A6 the need to reduce testing time and the benefits to be gained are discussed in greater detail. The project was to investigate how the time spent in thermal performance testing heat exchangers could be reduced.

The first phase of the project was to decide how testing time could be cut. Two approaches considered were the possible adoption of a new test method and the speeding up of present steady state testing by closed-loop control. Since the requirement was the testing of industrial heat exchangers with the appropriate working fluids, new test techniques were confined to those which met this constraint. Therefore, methods of testing by analogy, for example, mass transfer, were not included. The project concentrates on the heat transfer aspects of performance testing, and although of equal importance, the determination of pressure loss characteristics is not considered.

To approach the problem of possible new testing methods the need for steady state conditions when testing heat exchangers was questioned. Would it be possible to develop a non-steady state, i.e. a transient method, to determine heat exchanger performance? Non-steady state testing has the potential to reduce testing time, because the need for a long period of time to allow conditions to stabilize would be eliminated. A state-of-the-art review of non-steady state methods of testing heat exchangers was conducted, and the possibility and attractiveness of applying them to the company's products assessed. This forms Chapter One.

In short, none of the test techniques reviewed appeared attractive and consequently a new and novel approach to testing was proposed. This method, which this report starts to investigate, is hereafter called the 'dynamic test method.' The principle of the dynamic test method comes from the field of control engineering and is introduced in Chapter Two.

To assess reliably the dynamic test method, it was necessary to establish reference performance curves by the conventional steady state method, for both the water to water and oil to water fluid combinations. This forms Chapter Three.

Chapter Four details the experimental work conducted on the dynamic test method.

Chapter Five gives the analysis and discussion of the dynamic test results. Unfortunately, time did not permit a detailed analysis of the results and consequently the suitability of dynamic testing as an alternative testing method has not been fully evaluated.

The second approach to reducing test time, that of closed-loop control of steady state testing, is presented in Appendix A6.

Conclusions and Recommendations follow Chapter Five.

I.3.1. FUNDAMENTALS OF HEAT EXCHANGER THERMAL PERFORMANCE

In this recap of the basic heat transfer aspects used in the determination of heat exchanger performance the term heat exchanger refers to those of the recuperative type, where two fluids which are exchanging heat are separated by a heat conducting wall.

Newton defined the rate of heat transfer from the surface of a solid to a fluid flowing over it by the equation:

$$\dot{Q} = hA(\theta_w - \theta) \tag{I.1}$$

Inherent in this statement is the assumption that the heat transfer coefficient, which includes the effect of both convection and conduction to the fluid, is a mean value applicable to the heat transfer surface area and implies that there is no appreciable variation in its value over the surface. Further, the heat transfer due to radiation is not included. Radiation can be calculated separately, however, unless the surface temperature is high, or the surface loses heat by natural convection, then its effect can be neglected.

Where the variation in the heat transfer coefficient needs to be taken into account it is useful to use the concept of a local heat transfer coefficient defined by:

$$\frac{d\dot{Q}}{dA} = h_x(\theta_w - \theta) \tag{I.2}$$

although this is more difficult to measure than the average heat transfer coefficient. An average coefficient may then be obtained by integrating the local heat transfer coefficient over the entire heat transfer surface area.

In the case of a two fluid heat exchanger it is customary to use an average overall heat transfer coefficient based on the overall effective temperature difference (see later) between the fluids and is given by:

$$U = \frac{1}{\frac{1}{h_o} + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right) + \frac{\Delta r}{k} + r_{of} + r_{if}\left(\frac{A_o}{A_i}\right)} \qquad (I.3)$$

where U is arbitrarily based on the outside surface area of the tubes and any over tube secondary surface. Equation I.3 reduces to:

$$U = \frac{1}{\frac{1}{h_o} + \frac{1}{h_i}\left(\frac{d_o}{d_i}\right)} \qquad (I.4)$$

provided that:

(a) There is no secondary (extended) surface fitted.

(b) The fouling resistance is negligible in comparison with the resistance to heat transfer offered by the boundary layer film, which is true for "clean" heat exchangers.

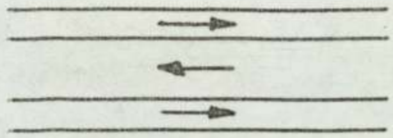
(c) The thickness of the tube wall is small and its thermal conductivity large such that $\frac{\Delta r}{k} \rightarrow 0$

By considering the heat transfer over a small area a local overall heat transfer coefficient can also be obtained.

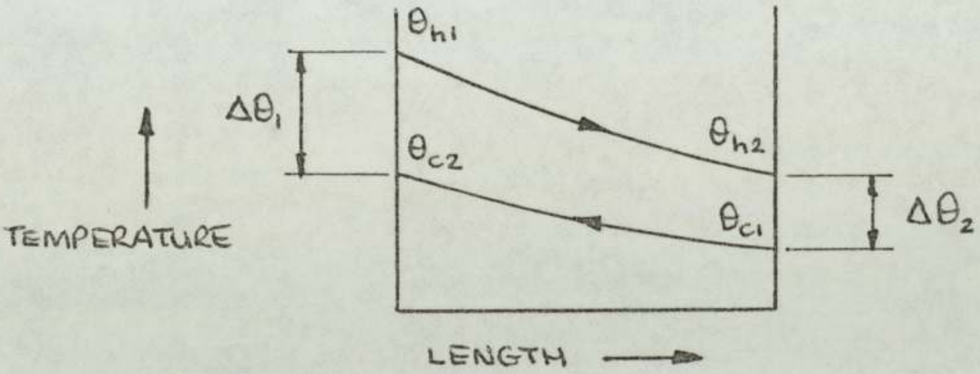
In a heat exchanger, since the temperature difference between the two fluids will generally vary along its length, it is convenient to use a logarithmic mean temperature difference, LMTD. For a pure counterflow or parallel flow heat exchanger, (fig. I.1), the LMTD is usually given by:

$$\Delta\theta_{LMTD} = \frac{\Delta\theta_1 - \Delta\theta_2}{\ln \frac{\Delta\theta_1}{\Delta\theta_2}} \qquad (I.5)$$

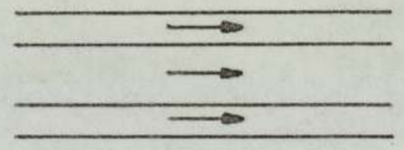
In the derivation of this equation for the effective temperature difference it is assumed that:



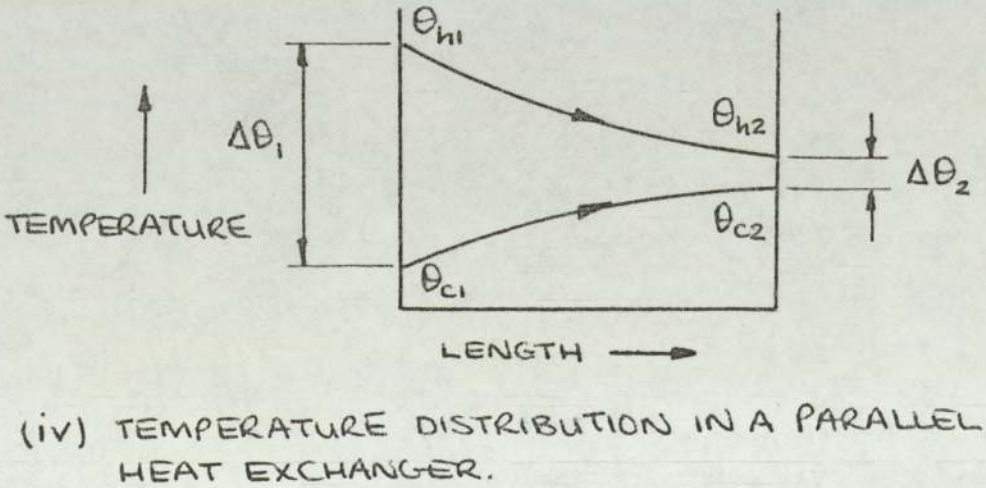
(i) PURE COUNTERFLOW



(ii) TEMPERATURE DISTRIBUTION IN A COUNTERFLOW HEAT EXCHANGER.



(iii) PURE PARALLELFLOW



(iv) TEMPERATURE DISTRIBUTION IN A PARALLELFLOW HEAT EXCHANGER.

FIG. I.1 PURE COUNTERFLOW AND PARALLELFLOW ARRANGEMENTS AND TEMPERATURE DISTRIBUTIONS.

(a) The overall heat transfer coefficient is constant over the length of the heat exchanger.

(b) The temperature of the shellside fluid is uniform over any cross section. This implies complete mixing of the fluid.

(c) The specific heat at constant pressure of each fluid does not vary with temperature.

(d) The flow rate of each fluid is constant.

(e) There are no partial phase changes in the heat exchanger.

(f) Heat losses are negligible.

In general, flow through a heat exchanger will not be pure counterflow or parallel flow (except in the case of the double pipe heat exchanger) but will consist of counterflow and/or parallel flow components plus crossflow components giving a mixed flow arrangement. There is often a significant difference between the LMTD for a pure counterflow heat exchanger and the LMTD for a pure crossflow unit given the same inlet and outlet temperatures. This is because pure cross flow results in non-uniform outlet temperatures (fig. I.2). For practical purposes the relationships for pure counterflow and parallel flow may be used for single tube pass shell and tube heat exchangers provided the shellside is well baffled. The baffles force the fluid to flow back and forth across the tubes many times as the fluid travels along the length of the exchanger, thus promoting good fluid mixing and approximating counter or parallel flow.

When the LMTD given by equation I.5 cannot be applied as in the case of units with two or more shell or tube passes it is necessary to introduce a correction factor such that the heat transfer rate is given by:

$$\dot{Q} = UA_o F_c \Delta\theta_{LMTD} \qquad (I.6)$$

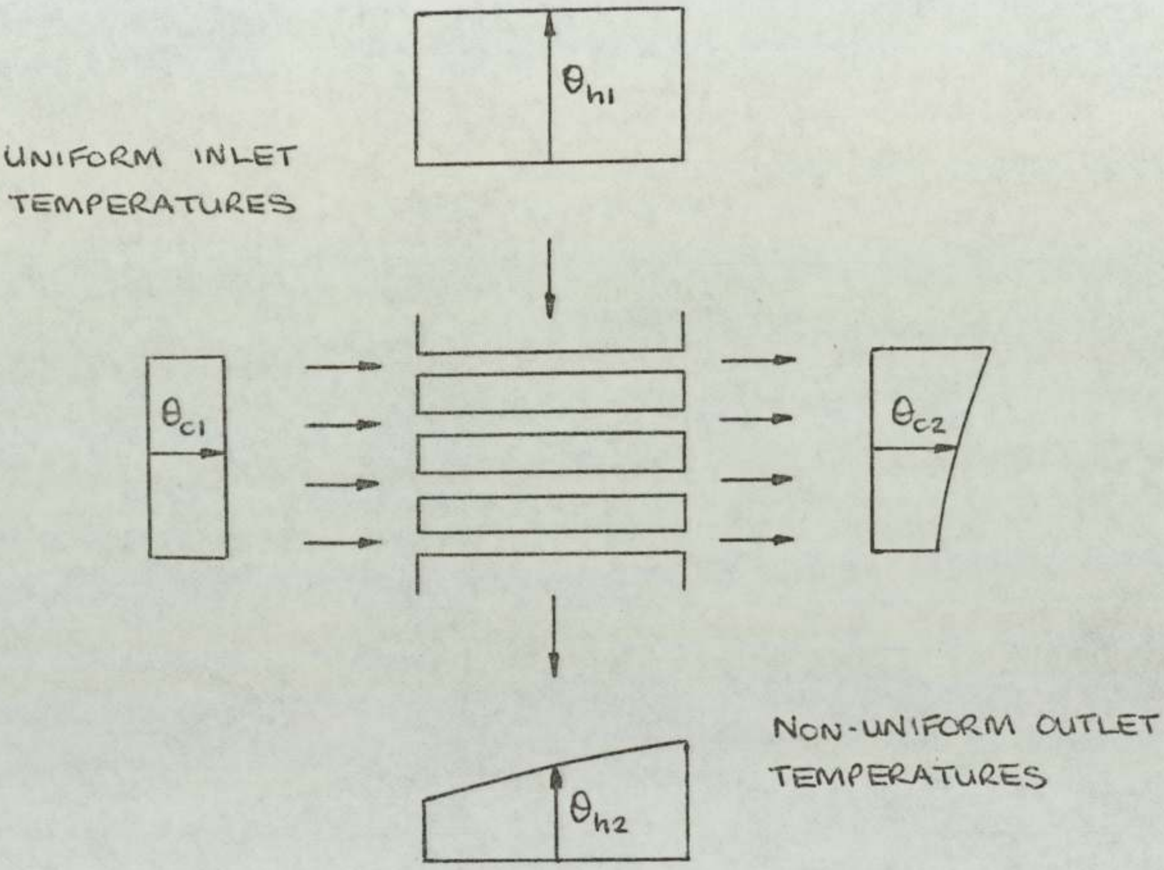


FIG. I.2 PURE CROSSFLOW ARRANGEMENT SHOWING TYPICAL INLET AND OUTLET TEMPERATURE DISTRIBUTIONS.

where F_c = the correction factor to be applied to the LMTD for counterflow should the heat exchanger be of multipass fluid arrangement or cross flow arrangement. F_c has a maximum value of 1 which is for counterflow. (See reference 6).

In addition to equation I.6 the equations:

$$\dot{Q} = \dot{m}_h c_h (\theta_{h1} - \theta_{h2}) \tag{I.7}$$

$$\dot{Q} = \dot{m}_c c_c (\theta_{c2} - \theta_{c1}) \tag{I.8}$$

expressing the heat transfer rate from and to each fluid are used in the determination of heat exchanger performance, provided heat losses are negligible.

The partial heat transfer coefficient is a function of the physical properties of the fluid, the geometry of the surface past which the fluid is flowing and between which heat is being transferred, and the velocity of the fluid relative to the surface. For all but the simplist of cases, the relationship between these parameters and the partial heat transfer coefficient as described by the mathematical equations of fluid mechanics and heat transfer are exceedingly complex and difficult to solve. Consequently, the relationship is generally determined empirically.

The thermal resistance to heat transfer at the fluid/surface interface is confined to a comparatively thin boundary layer through which heat is transferred predominantly by conduction. Thus, if the thickness of this boundary layer is δ, the rate of heat transfer through it is given according to Fourier's Law for one dimensional heat conduction as:

$$\dot{Q} = \frac{kA}{\delta} (\theta_w - \theta) \tag{I.9}$$

comparing equations I.1 & I.9 : $h = \dfrac{k}{\delta}$ (I.10) and therefore, one method of finding a value for the heat transfer coefficient is to measure the thickness of the boundary layer. However, it is difficult to measure the thickness of the fluid film or the temperature at the interface between the film and the main bulk of the fluid.

Equation I.1 shows that the partial heat transfer coefficient can be determined directly by dividing the heat transfer rate per unit area by the temperature difference between the surface and the bulk of the fluid. In experimental work on double pipe heat exchangers it is possible to measure this temperature difference, but in most industrial heat exchangers their construction usually makes these measurements impractical. The difficulty then arises in separating the two partial heat transfer coefficients from the overall coefficient. One method is to arrange that one partial heat transfer coefficient is very much greater in value than the partial coefficient to be determined, thus, any reasonable error in its estimation will have a negligible affect on the overall coefficient and hence on the derived partial coefficient. In other words, the partial heat transfer coefficient to be determined is arranged to be the one controlling the overall coefficient since the smaller coefficient provides the major resistance to heat transfer. Then, depending on whether it is necessary or not to include the thermal resistance of the tube wall or an allowance for fouling, equation I.3 or I.4 is used to extract the required partial heat transfer coefficient. A second method is a graphical procedure due to Wilson (53) and called the Wilson Plot Technique. In this technique the fluid flow rate is varied on one side of the heat exchanger only and the resulting variation in the overall heat transfer coefficient is assumed to be a function of that side's partial coefficient.

To illustrate the principle, equation I.3 is rewritten, assuming no secondary surface is fitted and zero tube resistance, as:

$$\frac{1}{U} = \frac{1}{h_o} + \frac{1}{h_{ir}} + r_{of} + r_{irf} \qquad (I.11)$$

where $\frac{1}{h_{ir}} = \frac{1}{h_i}\left(\frac{d_o}{d_i}\right)$ = inside tube heat transfer coefficient referred to the tube outside surface.

$r_{irf} = r_{if}\left(\frac{d_o}{d_i}\right)$

If the overtube partial heat transfer coefficient is desired, then the referred through tube coefficient is assumed to follow the law:

$$h_{ir} = B\dot{m}^n \qquad (I.12)$$

and equation I.11 becomes:

$$\frac{1}{U} = A + \frac{1}{B\dot{m}^n} \qquad (I.13)$$

where $A = \frac{1}{h_o} + r_{of} + r_{irf}$, which is essentially constant for the tests.

B = the non-varying factors in the film coefficient.

A graphical plot, fig. I.3, of $1/\dot{m}^n$ versus $1/U$ will produce a straight line of slope $1/B$ and intercept A. If the two fouling resistances are negligible, then A becomes the overtube film resistance, (the reciprocal of the overtube heat transfer coefficient). The difficulties met when trying to use the Wilson Plot Technique are:

1. The relationship between the partial heat transfer coefficient and the mass flow rate must be correctly assumed. The value of the flow exponent, n, may vary during the tests, for example, should the flow pass from the laminar regime through to the turbulent regime.

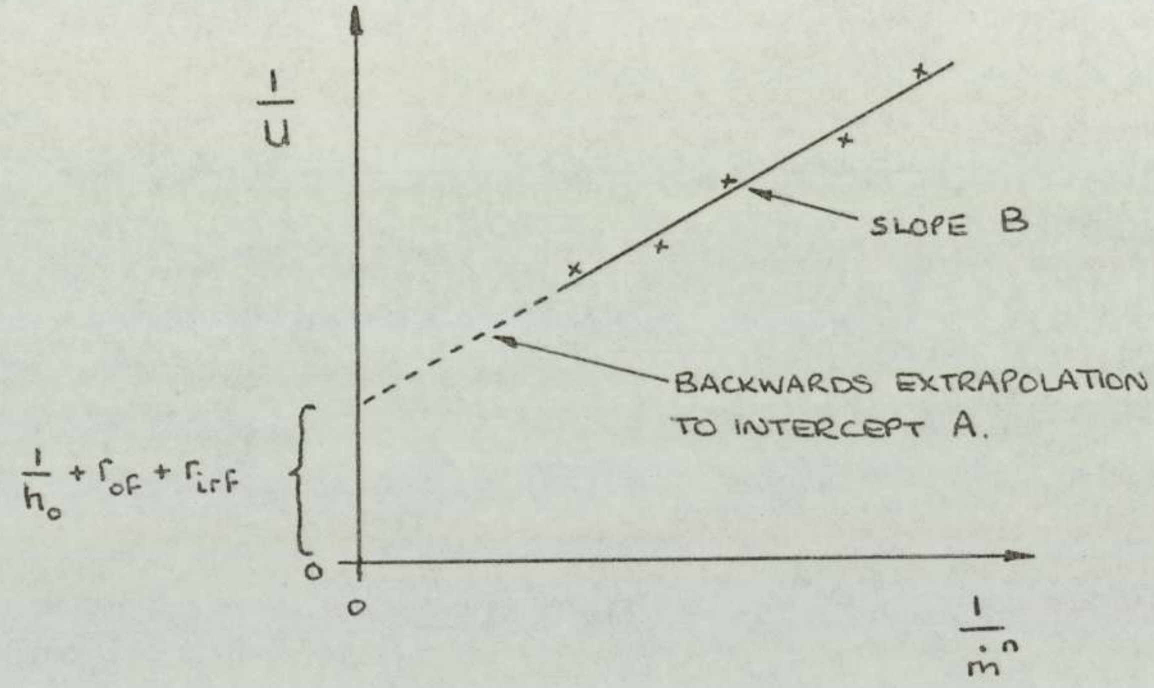


FIG. I.3 THE WILSON PLOT TECHNIQUE

2. There may be a significant variation in the temperatures during a complete test run and therefore, also in the physical properties of the fluids, thus introducing errors into the results obtained. This problem can be reduced if the mean bulk fluid temperatures on both sides of the heat exchanger are maintained constant, however, this makes the data troublesome to collect.
3. The accurate location of the intercept A. This requires a large number of data points and at high flow rates (corresponding to low values of $1/\dot{m}^n$) the accuracy of the calculated overall heat transfer coefficient is dependant on the accuracy with which small temperature differences can be measured. Thus, an error in the slope B will often produce a large error in the partial heat transfer coefficient when determined from the intercept due to the backwards extrapolation of the line.

Briggs & Young (7) have developed a modified Wilson Plot Technique for obtaining heat transfer correlations for shell and tube heat exchangers. The method accepts data collected at various flow rates and temperatures on either side of the heat exchanger and processes the data using a nonlinear regression procedure.

Dimensionless correlations for forced convective heat transfer incorporating the partial heat transfer coefficient usually take the form:

$$Nu.\left(\frac{\mu}{\mu_w}\right)^{-0.14} = f_1\ (Re,\ Pr,\ L/D) \qquad (I.14)$$

$$\text{and } j_h' = St.Pr^{\frac{2}{3}}\left(\frac{\mu}{\mu_w}\right)^{-0.14} = f_2\ (Re,\ L/D) \qquad (I.15)$$

where $\left(\frac{\mu}{\mu_w}\right)^{-0.14}$ = viscosity correction term allowing for significant viscosity changes near the tube surface.

For the purpose of such correlations it is necessary to make allowance for the variation in fluid properties with temperature. For moderate temperature differences across a heat exchanger the properties are evaluated at some mean temperature, often the bulk mean temperature, $\theta_b = (\theta_1 + \theta_2)/2$ or the fluid film temperature, $\theta_f = (\theta_w + \theta_b)/2$. (In the viscosity correction term in equations I.14 & I.15, μ_w is the viscosity of the fluid evaluated at the tube wall temperature). The partial heat transfer coefficient calculated from these correlations is therefore a mean coefficient.

For a more detailed coverage of the heat transfer aspects used in calculating heat exchanger performance the reader is referred to texts such as references 15, 25, 26 & 36.

I.3.2. PERFORMANCE TESTING OF HEAT EXCHANGERS AND THE INFORMATION DERIVED FROM THESE TESTS.

Heat transfer testing of heat exchangers is traditionally carried out under steady state conditions. For the purpose of this report steady state testing is defined as the condition whereby temperatures do not change with time but may vary with axial position and where flow rates do not change with time. Expressed mathematically:

$$\left.\frac{\partial \theta}{\partial t}\right|_{x\,=\,\text{constant}} = 0 \qquad \text{and} \quad \frac{\partial \dot{m}}{\partial t} = 0 \qquad \text{(I.16)}$$

For consistent and repeatable data it is essential that these conditions, or as near to them as is practically possible, be attained. In order to achieve a steady state condition it is necessary to allow the heat transfer test rig and heat exchanger to stabilize thermally. Depending on the size of the test rig and test unit and the associated heat capacities this may take any time between 15 minutes and several hours using current test facilities. Particular problems in obtaining temperature stabilization are fluctuations in water supply temperature and the demand for process steam for heating. Then, even with control equipment these disturbances can upset the thermal equilibrium required prior to and during the recording of test data.

The prime source of errors in heat transfer tests is in temperature measurement, particularly when small temperature differences are involved. If, for example, the temperature difference between the inlet and outlet of a heat exchanger for one fluid is only 1K, then an error in its determination of 0.1K will produce a 10% error in the calculated heat transfer (heat dissipation) rate, assuming the mass flow rate and specific heat to be correct. This error would result if the temperature difference is obtained by subtracting two thermometer readings that are incorrect by + 0.05 and - 0.05K. It is essential to ensure complete fluid mixing at the temperature sensor so that a single bulk (or mixing cup) temperature is obtained. Any temperature gradients existing in the fluid can result in significant temperature measurement errors. This is particularly important when the fluid flow is likely to be laminar because significant temperature gradients will be established across the fluid and it will become essential to create a high degree of turbulence to promote mixing. Also, this is important at the heat exchanger outlets where fluid particles having traversed different paths through the heat exchanger and emerging at different temperatures will come together to produce a temperature variation within the fluid.

A very important tool in heat transfer testing is the heat balance. It is obtained by comparing the heat given up by the hot fluid with the heat gained by the cold fluid (equations I.7 & I.8) and the estimated heat losses. The heat balance is often expressed as a percentage and if heat losses are negligible, as should be the case for a well lagged heat exchanger, may be written as:

$$\text{heat balance} = \left[\frac{\dot{m}_c c_c(\theta_{c2} - \theta_{c1})}{\dot{m}_h c_h(\theta_{h1} - \theta_{h2})} - 1\right] \times 100 \ \% \qquad (I.17)$$

Any disparity in the heat balance can be examined for an indication as to the probable source of error. This will generally be due to flow or temperature measurement, or lack of temperature stabilization of the test rig and may be associated with the manner in which the testing is conducted and the conditions changed, for example, flows and temperatures raised or lowered.

There are four main reasons for thermal performance testing a heat exchanger:

1. To provide fundamental heat transfer and pressure loss data.
2. For development work using prototypes.
3. To demonstrate to a customer that a particular heat exchanger will meet the specified duty requirements.
4. For quality control.

To provide basic design data and assist in development work, tests are usually carried out over a wide range of flow rates and temperatures to give a wide variation in Reynolds Number. The partial heat transfer coefficients are then derived and presented in non-dimensional plots. Pressure loss data is usually based on isothermal tests and is presented in non-dimensional form as Reynolds Number versus friction factor. The friction factor is deduced after making suitable allowances for abrupt enlargements and contractions etc. in the flow passages and represents the loss due to skin friction and form drag.

The presentation of results for a customer may vary, but often takes the form of a carpet plot of shell and tube fluid flow rates versus the ratio, $Q/(\theta_{h1} - \theta_{c1})$. This graph enables the user of the piece of equipment on which the heat exchanger is fitted to make a quick estimate of the heat being dissipated by the heat exchanger given only the two inlet temperatures and the flow rates. The graph is presented for a series of flow rates above and below the design condition and shows the user the performance of the heat exchanger away from the design point. Occasionally, the flow rates are plotted against the thermal ratio, $(\Delta\theta_h$ or $\Delta\theta_c)/(\theta_{h1} - \theta_{c1})$, which is a measure of the thermodynamic performance of the heat exchanger and has a maximum value of unity. Isothermal pressure loss data is presented as flow rate versus the total pressure drop across the heat exchanger for each fluid at the appropriate temperatures.

Regular thermal performance testing heat exchangers for quality control purposes, although perhaps desirable, is not practical at present while heat transfer tests take such a long time to perform. Certain heat exchangers are flow tested as a quality control measure to ensure satisfactory thermal performance. Here, a fixed differential pressure is applied across the heat exchanger and the resulting flow measured. Provided the flow is within set tolerances the heat exchanger is accepted or rejected. In the latter case the reason for its quality failure is traced. Flow testing is used because both the flow rate (for a given pressure drop) and the heat transfer rate are particularly sensitive to fluid by-passing. By-passing of the fluid generally occurs on the shellside of a unit and, if excessive, will result in a marked decrease in the heat transfer performance of the heat exchanger and will be accompanied by a decrease in the pressure loss across the shellside. By-passing is caused by the clearances, necessary for manufacture, between surfaces which are in close proximity to one another. Any increase in the size of the clearance will lower the resistance to flow in this region and part of the flow will take this path of lower resistance and will not flow across the whole of the heat transfer surface, therefore, reducing the heat exchanger efficiency.

CHAPTER ONE

REVIEW AND ASSESSMENT OF EXISTING TESTING TECHNIQUES

1.1. REVIEW OF TESTING TECHNIQUES

As has been stated in the introduction to this report, there is an indication that performance testing time would be reduced by the adoption of a non-steady state, i.e., transient, test technique which would give results comparable with those from the existing steady state method. This review covers transient type methods which have been used to obtain the heat transfer characteristics of surfaces for heat exchangers and the heat exchanger units themselves.

1.1.1. THE THERMAL CAPACITOR CYLINDER METHOD

An early use of a transient test technique was reported by Kays, London, and Lo (24) to obtain the heat transfer characteristics for air flow normal to tubebanks. The principle on which the method is based is by considering the heat transfer between a small body immersed in a fluid stream and the stream. The method required only one fluid stream. A thermal capacity cylinder heated to above ambient was inserted into the tubebank in place of one of the tubes and ambient air blown through the test core. The temperature-time history of the heat capacitor was recorded and from a simple graphical plot the local heat transfer coefficient determined. The heat transfer coefficient is related to the temperature-time history by the equation:

$$\frac{\theta_s - \theta_o}{\theta_{si} - \theta_o} = e^{\frac{-hAt}{W_s c_s}} \qquad (1.1)$$

They stated that for staggered tubebank arrangements it was believed that mixing of the air was sufficient to enable data determined for a single tube near the centre of the tubestack to be used to predict the performance of the whole tubestack. In addition, it has been found for viscous main stream flow that there is a significant difference between transient single tube results and data from whole tubebanks tested under steady state. It was concluded that the transient technique gave results in good agreement with results obtained under steady state conditions provided the flow was fully turbulent.

1.1.2. THE SINGLE-BLOW TECHNIQUE

The single-blow technique utilises a single fluid, often air, and consists of imposing a step change in temperature on the fluid flowing into the test core and measuring the temperature response of the fluid flowing out of the core. The single blow technique is described by two partial differential equations:

for the surface:

$$\rho_s A_s c_s \frac{\partial \theta_s}{\partial t} = \frac{hA}{L}(\theta_f - \theta_s) \qquad (1.2)$$

and for the fluid:

$$\dot{m}_f c_f \frac{\partial \theta_f}{\partial x} = \frac{hA}{L}(\theta_s - \theta_f) \qquad (1.3)$$

Introducing a generalised time variable,

$$t^* = \frac{hA}{W_s c_s} t$$

and a generalised position variable,

$$z = \frac{hA}{\dot{m}_f c_f} \cdot \frac{x}{L} = NTU \cdot \frac{x}{L}$$

then, making these substitutions, equations 1.2 and 1.3 become:

$$\frac{\partial \theta_s}{\partial t^*} = \theta_f - \theta_s \tag{1.4}$$

$$\frac{\partial \theta_f}{\partial z} = \theta_s - \theta_f \tag{1.5}$$

In 1929 Schumann (47) obtained the analytical solution to equations 1.4 and 1.5 as:

$$\frac{\theta_f - \theta_i}{\theta_{f1} - \theta_i} = 1 - e^{-(z+t^*)} \sum_{n=1}^{\infty} X\,(t^*, z) \tag{1.6}$$

$$\frac{\theta_s - \theta_i}{\theta_{f1} - \theta_i} = 1 - e^{-(z+t^*)} \sum_{n=0}^{\infty} X\,(t^*, z) \tag{1.7}$$

where $X(t^*,z) \;= z^n \dfrac{d^n}{d(zt^*)^n} \left(J_o(2j\sqrt{t^*.z})\right)$

J_o = zero order modified Bessel function of the first kind.

$j = \sqrt{-1}$

When $x = L$, $\theta_f = \theta_{f2}$ and $z = NTU$, then, by matching the experimentally obtained curves of the fluid outlet temperature with Schumann's theoretical curves, (the family of curves represented by equation 1.6 for constant values of NTU), heat transfer coefficients can be determined. This technique was first used by Furnas (16) in the early 1930's.

The next significant development in the technique came in 1950 when Locke (31) differentiated Schumann's theoretical solution and obtained the maximum slope of the outlet temperature response curve as a unique function of the NTU:

$$\frac{d\left[\dfrac{\theta_{f2} - \theta_i}{\theta_{f1} - \theta_i}\right]}{d\left[\dfrac{t^*}{NTU}\right]} = \frac{NTU^2}{\sqrt{NTU.t^*}}\left[-jJ_1(2j\sqrt{NTU.*t})\right] e^{-(NTU + t^*)} \tag{1.8}$$

where J_1 = first order modified Bessel Function of the first kind.

This relationship is shown in fig. 1.1 for $\lambda=0$, (see figure for definition of λ). The method which has become known as the maximum slope method gives smaller uncertainties in the test results than other methods of data evaluation. However, the method is limited to NTU$>$3.5 because a point of inflex ion occurs at NTU=2 in the NTU versus maximum slope curve which results in the NTU being very sensitive to changes in the value of the maximum slope.

Schumann's solution is based on the assumption that longitudinal thermal conduction in the test matrix material is zero. This assumption can give rise to serious errors as shown by Mondt (39) when he compared the effect of zero, infinite and a few intermediate values of finite longitudinal conduction on the NTU - maximum slope curve. A year later, in 1964, Howard (23) using the finite difference technique, presented in tabular and graphical form a more comprehensive treatment of the effect of longitudinal thermal conduction. Fig. 1.1 shows the NTU as a function of maximum slope and the longitudinal conduction parameter, λ.

Kohlamyar (27,28) has extended the maximum slope method to include arbitary upstream fluid temperature changes. This enables a theoretical correction factor to be applied which accounts for the imposed step input in temperature not being a perfect step.

To overcome the problem of the lower limiting value of NTU for the application of the maximum slope method several techniques have been applied. Mondt and Siegla (40) have used a procedure which has become known as the "initial rise"

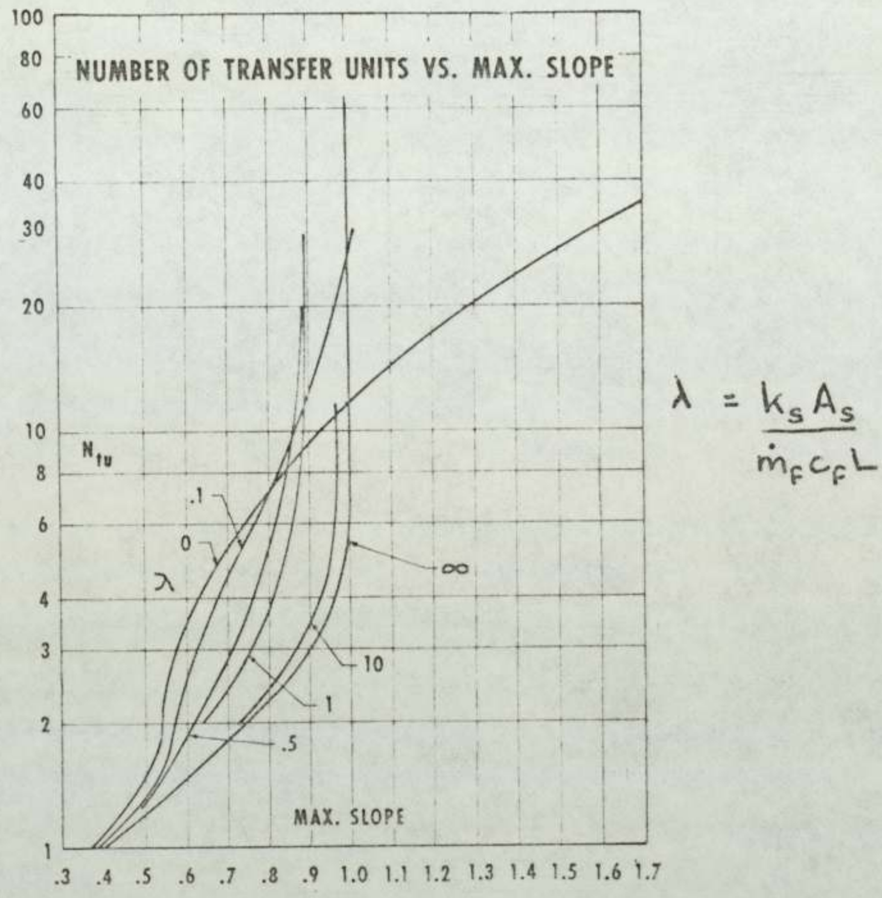


FIG. 1.1 NUMBER OF HEAT TRANSFER UNITS AS A FUNCTION OF MAXIMUM SLOPE AND LONGITUDIAL CONDUCTION PARAMETER, λ.

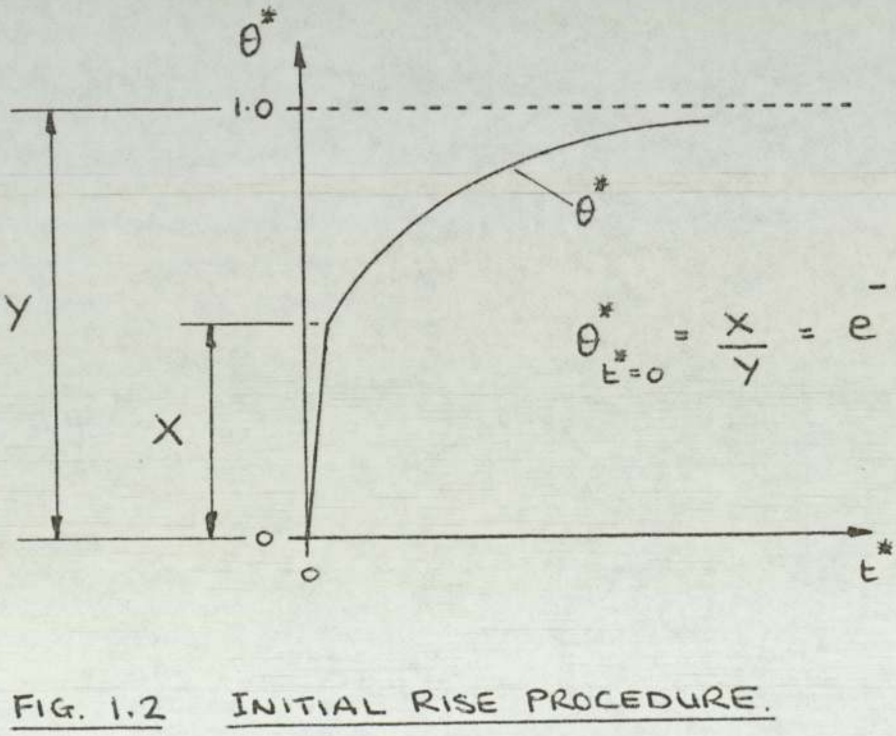


FIG. 1.2 INITIAL RISE PROCEDURE.

procedure. This procedure is based on the assumption that the response of the exit fluid temperature is a fractional step rise occurring at zero time, followed immediately by an exponential rise - fig. 1.2. Substitution of $t^* = 0$ and $x = L$ into equation 1.6 gives:

$$\theta^*_{f2} = \frac{\theta_{f2} - \theta_i}{\theta_{f1} - \theta_i} = \frac{X}{Y} = e^{-NTU} \tag{1.9}$$

Mondt and Siegla also state that this method is insensitive to longitudinal conduction effects.

In 1975 Liang and Yang (29) proposed their modified single blow technique. They pointed out that at low values of NTU the accuracy of the maximum slope method depends strongly on how close the actual inlet temperature change is to a step input. In their analysis they assumed an exponential inlet temperature rise and demonstrated its validity; thus, the boundary condition at inlet to the test section is defined by:

$$\theta^*_f\ (t^*, 0) = 1 - e^{\frac{-t^*}{T^*}} \tag{1.10}$$

where $T^* = \frac{hAT}{W_s c_s}$, dimensionless

T = time constant of the inlet temperature, s

Their mathematical model differed from previous models in that it included the term for the heat storage capacity of the fluid; the term for longitudinal conduction was ignored. Solution of their model was obtained by Laplace Transform techniques and resulted in the outlet fluid temperature being given by the equation:

$$\theta^*_{f2}(t^*, NTU) = \frac{1}{T^*}\int_{\beta^*}^{t^*} e^{-(t^*-\eta)/T^* - b_2\beta} \times \left\{ e^{-(\eta-\beta^*)} J_0\left[2\sqrt{b_2 \beta^*(\eta-\beta^*)}\right] + \Phi(\eta, NTU)\right\} d\eta \tag{1.11}$$

$$\text{where} \quad \Phi(\eta, NTU) = \int_0^{\eta-\beta^*} e^{-\xi} J_0\left[2(b_2\beta^*\xi)^{1/2}\right] d\xi$$

η, ξ = dummy variables

β^* = NTU/b_1, dimensionless

b_1 = $(V_f/A_c)b_2$, dimensionless

b_2 = $W_s c_s/W_f c_f$, dimensionless

V_f = volume of fluid in test core per unit length, m^2

A_c = core minimum free flow area, m^2

Average heat transfer coefficients were obtained by matching this expression to the exit air temperature response curve using a digital computer. This technique can be applied for any values of NTU.

Pucci, Howard and Piersall (43) give a good summary of the underlying theory to the single blow technique and the maximum slope method. Their paper also includes a description of the experimental rig and gives the results for a series of surfaces.

1.1.3. THE CYCLIC OR PERIODIC METHOD

The principle of the cyclic or periodic method is that a cyclic, usually sinusoidal, variation in temperature is imposed on the upstream fluid. Heat is exchanged between the fluid and the test surface as the fluid passes over the surface and results in an attenuation in the amplitude ratio of the fluid inlet to outlet temperature and a corresponding phase shift. The measurement of either the temperature amplitude ratio or the phase shift, plus details of the surface geometry, the physical properties of the fluid and surface material can, together, be related to the convective heat transfer coefficient. This technique requires only one working fluid.

The amplitude ratio can usually be measured more accurately and conveniently than the phase difference; consequently the amplitude ratio is generally used in the calculation of the heat transfer coefficient.

The development of the cyclic method stems from the initial work of Bell and Katz (4) reported in 1949. They presented the solution to the problem for three cases:

(a) Infinite thermal conductivity in the test core material.

(b) The thermal conductivity in the test core material is zero in the longitudinal (flow) direction and infinite in the transverse direction.

(c) The thermal conductivity in the test core material is zero in the longitudinal direction and finite in the transverse direction.

For each of the above cases respectively, the amplitude ratio R, of the inlet temperature to the outlet temperature is:

(a) $$R = \left[\frac{e^{-2.NTU} + M^2\ (1 - e^{-NTU})^2}{1 + M^2\ (1 - e^{-NTU})^2}\right]^{\frac{1}{2}} \tag{1.12}$$

(b) $$R = \exp\left[-\frac{NTU}{1 + (M.NTU)^2}\right] \tag{1.13}$$

(c) $$R = \exp\left\{-\frac{NTU}{1 + (M.NTU)^2}\left[1 + \frac{\sigma^2 M.NTU(M^2 NTU^2 - 1)}{6(1 + M^2 NTU^2)} + \ldots\right]\right\} \tag{1.14}$$

where $M = \dfrac{\dot{m}_f c_f}{W_s c_s \omega}$, dimensionless

$\sigma^2 = \dfrac{\omega \delta W_s c_s \rho_s}{2k_s}$, kg/m (see page 41a)

δ = plate thickness, m

Equations 1.12 to 1.14 can then be solved for the heat transfer coefficient.

In 1952 Dayton etal (12) extended the theory to include test surfaces of low thermal conductivity and give the amplitude ratio as:

$$R = \exp\left\{ - \left[\frac{nq + nq^2}{1 + 2q + 2q^2} \right] \right\} \tag{1.15}$$

where $n = \dfrac{A\sqrt{2k_s c_s \rho_s \omega}}{A_f \rho_f \ c_f V}$, dimensionless

$q = \dfrac{h}{\sqrt{2k_s c_s \rho_s}}$, dimensionless

Their report is concerned chiefly with developing and proving the validity of the cyclic method and includes a useful evaluation of the errors introduced by the experimental conditions not satisfying the theoretical requirements.

Dingee and Chastain (13) in 1957 made a further study of the errors associated with the cyclic method. Their results indicated that most of the errors arising can be attributed to non-uniform flow distribution which results in mixing of the fluid and a further attenuation of the outlet temperature response. They state that this effect can be expected to be small if the fluid is air but not necessarily if it is water. They show that the accuracy of the technique is particularly sensitive to high density fluids and, therefore, is not suitable for tests involving fluids such as water - a small error in the measured amplitude ratio producing a large variation in the heat transfer coefficient.

Hart and Szomanski (19) give a short review of the theory of surface heat transfer measurement using the cyclic method and discuss the practical application of the theory.

When discussing its application to liquids they say that it appears that the effect of longitudinal conduction in the liquid must not be ignored. They also compare the cyclic method with the single blow technique, concluding that the cyclic method is preferable because of the greater precision with which the amplitude ratio can be measured and the flexibility in the choice of experimental parameters which the technique permits.

Pucci, Ball and Traister (44) have obtained heat transfer data for several surfaces by both the cyclic method and the single blow technique.

Stang and Bush (50) and Stang (51) have further developed the technique for the range $0.2 < NTU < 50$. Their mathematical model neglects the fluid thermal capacity relative to the thermal capacity of the test core material. This was considered in (50) and shown to be negligible when the fluid was a gas. Their model also includes the effect of longitudinal conduction in the core material and is described by the following equations:

$$\theta_f' - \theta_s' = \frac{\partial \theta_s'}{\partial t^*} - \lambda . NTU \frac{\partial^2 \theta_s'}{\partial z^2} \tag{1.16}$$

$$\theta_f' - \theta_s' = -\frac{\partial \theta_f'}{\partial z} \tag{1.17}$$

which, when subject to the boundary condition:

$$\theta_f'(0, t^*) = \sin \frac{t^*}{\psi} \tag{1.18}$$

where $\psi = \frac{1}{2\pi} \frac{t_o hA}{W_s c_s}$ and t_o = period of temperature oscillation.

yields the solution for the amplitude attenuation as:

$$R = e^{-\varepsilon_1 z} \tag{1.19}$$

and the phase shift as:

$$\phi = \varepsilon_2 z \tag{1.20}$$

where ε and ε_2 are functions of ψ and λ.NTU only and are given by:

$$\varepsilon_1 = \frac{\varepsilon_2}{\psi} - \lambda.\text{NTU}(-\varepsilon_1^{\,3} + \varepsilon_1^{\,2} + 3\varepsilon_1\varepsilon_2^{\,2}) \tag{1.21}$$

$$\varepsilon_2 = \frac{1-\varepsilon_1}{\psi} - \lambda.\text{NTU}(\varepsilon_2^{\,3} - 3\varepsilon_1^{\,2}\varepsilon_2 + 2\varepsilon_1\varepsilon_2) \tag{1.22}$$

Since the inlet and outlet temperature responses are not in general pure sine waves, a Fourier series analysis is used to extract the first harmonic. (Higher harmonics were found to contribute only a very small amount and therefore are not used in the heat transfer calculations). The first harmonic of the fluid inlet temperature can be represented by:

$$\theta_{f1}(0,t) = \frac{B_o}{2} + A \sin \frac{2\pi t}{t_o} + B \cos \frac{2\pi t}{t_o} \tag{1.23}$$

$$\text{where } A = \frac{2}{t_o} \int_0^{t_o} \theta_f(0,t) \sin \frac{2\pi t}{t_o}\, dt$$

$$B = \frac{2}{t_o} \int_0^{t_o} \theta_f(0,t) \cos \frac{2\pi t}{t_o}\, dt$$

$$\frac{B_o}{2} = \theta_m = \frac{1}{t_o} \int_0^{t_o} \theta_f(0,t)\, dt$$

and similarly for the fluid outlet temperature ($\frac{x}{L} = 1$) :

$$\theta_{f2}(1,t) = \frac{b_o}{2} + a \sin \frac{2\pi t}{t_o} + b \cos \frac{2\pi t}{t_o} \tag{1.24}$$

$$\text{where } a = \frac{2}{t_o} \int_0^{t_o} \theta_f(1,t) \sin \frac{2\pi t}{t_o}\, dt$$

$$b = \frac{2}{t_o} \int_0^{t_o} \theta_f(1,t) \cos \frac{2\pi t}{t_o}\, dt$$

$$b_o = \theta_m = \frac{1}{t_o} \int_0^{t_o} \theta_f(1,t)\, dt$$

The Fourier coefficients, A, B, a, b, are solved for numerically to determine the experimental amplitude attenuation and phase shift given by:

$$R_{expr} = \frac{\sqrt{a^2 + b^2}}{\sqrt{A^2 + B^2}} \qquad (1.25)$$

$$\phi_{expr} = \tan^{-1}\left[\frac{\frac{a}{b} - \frac{A}{B}}{1 + \frac{aA}{bB}}\right] \qquad (1.26)$$

The values from equation 1.25 and 1.26 are then substituted into equations 1.19 and 1.20 respectively and the heat transfer performance parameter NTU solved for by an iterative procedure. The largest source of error in the technique is considered to be that due to temperature measurement. To minimise the sensitivity of the technique to this uncertainty guidelines are given as to which data reduction procedure, amplitude attenuation or phase shift, should be used since the sensitivity of each method depends on the NTU value and the oscillation period of the input.

1.1.4. THE FREQUENCY RESPONSE METHOD

A variation on the cyclic or periodic method described above is the work reported in 1971 by Matulla and Orlicek (35). They used frequency response analysis to determine heat transfer coefficients in a double pipe, parallel flow, water-water heat exchanger. A sinusoidal temperature variation was applied at inlet to the inner tube of the exchanger and this temperature wave and the response wave at outlet from the outer tube recorded. This was repeated over a range of frequencies to produce an amplitude ratio - frequency plot or Bode diagram. As three heat transfer coefficients were involved - those of the inner fluid to inner tube; inner tube to outer fluid, and outer fluid to outer tube - the latter two heat transfer coefficients were assumed to be equal. The resulting pair of heat transfer coefficients were obtained by a regression method for which the sum of the square of the errors between the amplitude ratio - frequency curve predicted by their mathematical model and the experimental values was a minimum. The heat transfer coefficients fell within the range of the coefficients calculated from the literature used for the comparison. Their mathematical model consisted of four simultaneous partial differential equations given by:

for the inner fluid stream:

$$\rho_h \pi r_1^2 c_h \frac{\partial \theta_h}{\partial t} = -\dot{m}_h c_h \frac{\partial \theta_h}{\partial x} - 2\pi r_1 h_1 (\theta_h - \theta_{w1}) \qquad (1.27)$$

for the inner tube wall:

$$\rho_{w1} \pi (r_2^2 - r_1^2) c_{w1} \frac{\partial \theta_{w1}}{\partial t} = 2\pi r_1 h_1 (\theta_h - \theta_{w1}) - 2\pi r_2 h_2 (\theta_{w1} - \theta_c) \qquad (1.28)$$

for the outer fluid stream:

$$\rho_c \pi (r_3^2 - r_2^2) c_c \frac{\partial \theta_c}{\partial t} = -\dot{m}_c c_c \frac{\partial \theta_c}{\partial x} + 2\pi r_2 h_2 (\theta_{w1} - \theta_c) - 2\pi r_3 h_3 (\theta_c - \theta_{w2}) \qquad (1.29)$$

and for the outer tube wall:

$$\rho_{w2} \pi (r_4^2 - r_3^2) c_{w2} \frac{\partial \theta_{w2}}{\partial t} = 2\pi r_3 h_3 (\theta_c - \theta_{w2}) \qquad (1.30)$$

where the symbols, additional to those in the nomenclature, are defined in fig. 1.3. For counterflow operation the first term on the right-hand side of equation 1.27 becomes positive.

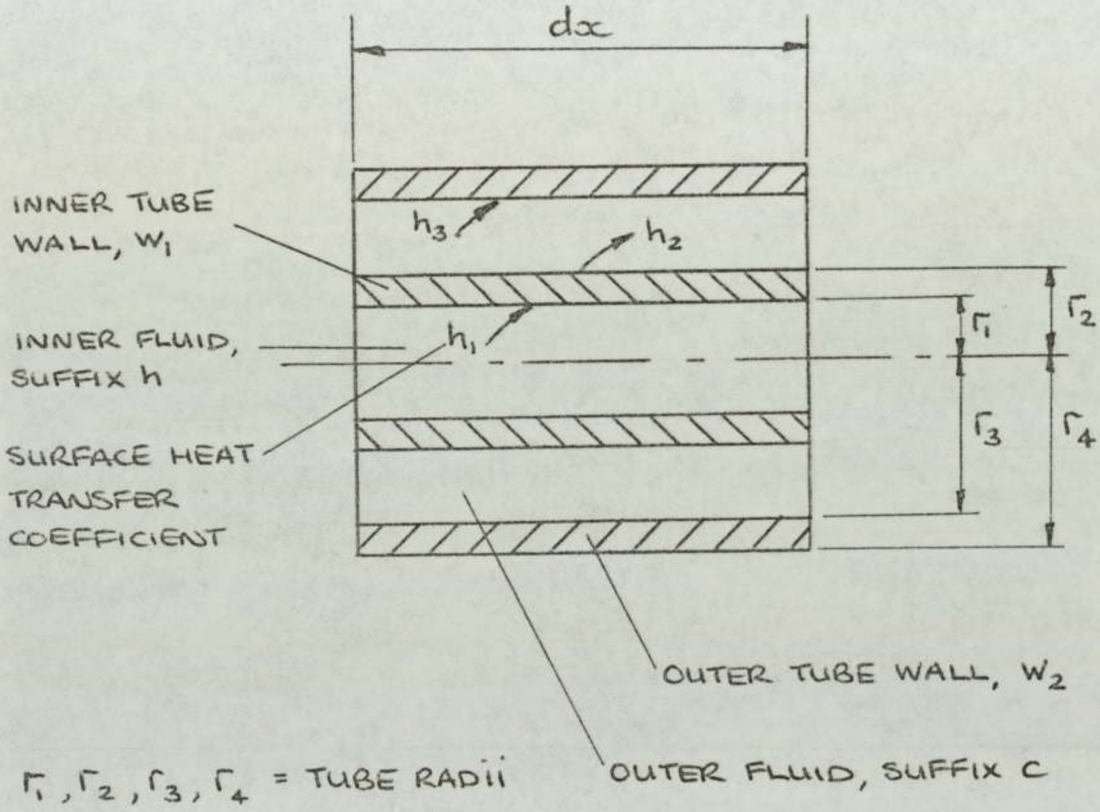


FIG. 1.3 DIFFERENTIAL ELEMENT OF DOUBLE PIPE HEAT EXCHANGER SHOWING MEANING OF SYMBOLS.

1.1.5. VARIATION ON THE SINGLE BLOW TECHNIQUE

In 1972 Gemza, Kotyk, and Komurka (17) determined partial and overall heat transfer coefficients for a double pipe, water-water heat exchanger from the transient response characteristics of one of the water streams following a step change in the stream's inlet temperature. The outer pipe of the heat exchanger was assumed to have zero thermal heat capacity and zero heat losses from it to the surroundings, therefore, eliminating a third heat transfer coefficient because, in their theory, no heat would be transferred to the outer pipe. Thus, their mathematical model consisted of equations 1.27, 1.28 and the first three terms of 1.29. In their analysis they considered the outlet temperature response to be composed of an initial step rise followed by a decaying exponential rise. The ratio of the experimental change in the outlet temperature from its initial steady state to the change in the inlet temperature from its initial steady state is related, via the authors mathematical model, to the heat transfer coefficients. From the initial step rise response the partial heat transfer coefficient on the side to which the temperature step is applied can be determined. The expression for the partial heat transfer coefficient is identical to that for the "initial rise" procedure of the single blow technique, equation 1.9. The overall heat transfer coefficient is determined from the final steady state change in temperature and is contained in the amplitude ratio:

$$\frac{\Delta\theta_2}{\Delta\theta_1} = \frac{1 - F}{e^{\frac{UA}{\dot{m}_c c_c}(1-F)} - F} \tag{1.31}$$

for counterflow, and

$$\frac{\Delta\theta_2}{\Delta\theta_1} = \frac{1}{1 + F}\left[F + e^{\frac{-UA}{\dot{m}_c c_c}(1+F)}\right] \tag{1.32}$$

for parallel flow arrangement, where

$\Delta\theta_1$ = change in steady state inlet temperature.

$\Delta\theta_2$ = change in steady state outlet temperature.

$$F = \frac{\dot{m}_c c_c}{\dot{m}_h c_h}$$

and the inequality $v_c > v_h$ must be fulfilled.
The other partial heat transfer coefficient can then be easily calculated.

1.1.6. USE OF A STATISTICAL METHOD

In 1973 Mumme and Lawther (41) using the results of Bell and Katz (4) developed a new transient type technique by considering random flow induced temperature fluctuations of a gas steam. They analysed the upstream and downstream temperature fluctuations by statistical methods using amplitude spectral density functions to determine the heat transfer characteristics of a tube bundle with air flow through the tubes. Only one fluid was used. For comparison purposes the heat transfer characteristics of the same test section was obtained using the cyclic method. The results of the two methods were in excellent agreement.

1.2. ASSESSMENT OF REVIEWED TESTING METHODS IN RELATION TO HEAT EXCHANGERS MANUFACTURED BY SERCK

It is not the intention of this assessment to repeat in detail points raised in the review of testing techniques. only to summarise and assess the possibilities of applying these techniques, or with modification, to thermal performance test heat exchangers manufactured by Serck.

Before proceeding with this assessment, it is useful to appreciate the background to the development of some of these techniques. The single blow technique was first applied to heat transfer tests on packed beds (16) on account of the problems associated with continuously heating or cooling the bed and measuring the temperature of the solids in steady state tests. The cyclic method of testing has also been applied to packed beds (37). With the development of gas turbines, particularly for vehicular applications, there has been an enormous amount of work carried out on compact heat exchanger surfaces (low volume, high surface density) for regenerator heat exchangers. Because steady state heat transfer tests are time consuming to perform and usually require two working fluids with the attendant problems of separating the partial heat transfer coefficients from the overall value, this necessitated the use of alternative experimental techniques which were relatively quick and reliable to perform. Both the single-blow and cyclic methods have been developed to test compact heat exchanger surfaces with the advantage that being single fluid testing methods the required heat transfer coefficient is obtained directly.

1.2.1. THE THERMAL CAPACITOR CYLINDER METHOD

The principal objections to transient testing using a heat capacity cylinder are that it is impractical for both shell and tube, and radiator type heat exchangers because there is no ease of access to place the cylinder within the exchanger, and in many cases the fluid would be a liquid. Further, it would probably be necessary to vary the position of the cylinder in at least two co-ordinates to determine the extent of the variation in local heat transfer coefficient and hence obtain an integrated overall value. However, it should be possible to extend the principle of the method to suitably designed models for use in development work but this course is unnecessary because work is already proceeding using the Limiting Diffusion Current Techniques (55), (a mass transfer analogy technique) to obtain performance information.

1.2.2. THE SINGLE-BLOW AND CYCLIC TECHNIQUES

These two techniques are assessed together because of the similarities of the two methods.

Both the single-blow and cyclic techniques have the advantages that as only one working fluid is required the the heat transfer coefficient for the surface being tested is obtained directly and there is no need to measure the surface temperature of the test matrix.

The single blow technique is dependant on the experimenter reproducing a step input. Fast response heater circuits have been developed in an attempt to reproduce the step input in temperature and the theory extended to take into account the true nature of the step input. With the cyclic or periodic method the temperature input is easier to generate. However, both techniques have been used to present data over the range $0.3<NTU<44$. Serck heat exchangers operate at the lower end of this range. Shell and tube units would not be expected to operate at NTU exceeding about 3.5, more frequently the upper limit of the NTU range would be between 1.5 and 2. Charge air coolers may have $NTU>5$ while radiators have a similar range to shell and tube units. In most analyses for the two techniques it has been assumed that the thermal capacity of the fluid was zero. This is a reasonable assumption for air (the test fluid) but cannot be expected to be valid for a liquid whose specific heat capacity is many times greater. It has also been shown that the cyclic method is particularly sensitive to high density fluids, thus making it unattractive for tests with liquids.

In the mathematical analysis of both techniques it was assumed that the differential element considered was representative of the matrix; this implies that the geometry of the matrix is uniform in the axial direction and over any cross section resulting in the steady flow of fluid with a flat velocity profile through the test matrix. From the literature it appears that matrices which have been tested by the two methods have no tubes or passages through which a second fluid could flow; in addition, the matrices were surrounded by an adiabatic surface. This means that only one heat transfer coefficient is involved in the analysis. However, in Serck heat exchangers there are normally two sets of fluid passages, one for the hot fluid and the other for the cold fluid, and an outer heat conducting casing. If, for example, a shell and tube heat exchanger is considered to be tested with a single fluid only, say over the tubes, then it may be necessary to consider two more heat transfer coefficients in addition to the coefficient required between the fluid and the tubes. These would be between the fluid and the shell and between the inner wall of the tubes and the relatively "stationary" air inside them.

1.2.3. THE FREQUENCY RESPONSE METHOD

The experimental disadvantage of the frequency response method discussed above is that it is necessary to cycle the temperature over a wide frequency range, (2 to 3 decades), resulting in numerous test points which are time consuming to carry out. This could be overcome by the use of the pulse testing method (22).

The assumptions inherent in the mathematical model are more likely to be experimentally satisfied in the simple case of the double pipe heat exchanger than in the more complex case of shell and tube units.

1.2.4. VARIATION ON THE SINGLE-BLOW METHOD

The comment made in section 1.2.3 concerning the mathematical model used to describe the process is also applicable to this test method.

The initial rise in temperature at the heat exchanger outlet following the step input is not a step as assumed in the analysis and does not exhibit a definite break point. It is necessary to estimate where the linear part of the response curve ends and use this to calculate the partial heat transfer coefficient. Thus, the magnitude of the coefficient determined is dependent on the accurate location of this estimated break point.

1.2.5. STATISTICAL METHOD

Statistical or stochastic methods are powerful analysis techniques usually only used when deterministic modelling techniques either do not give satisfactory results or are not applicable. Stochastic techniques are mathematically complex, increase the complexity of experimental equipment and require substantial computer facilities and, therefore, cannot be justified if simpler methods of analysis prove to be adequate. The testing of heat exchangers would normally be a controlled process without the introduction of random signals to excite the dynamics of the heat exchanger and consequently it is to be expected that it will not be necessary to resort to statistical methods of analysis.

1.2.6. GENERAL COMMENTS

In each of the above techniques the approach adopted in the mathematical analysis was to consider a typical differential element of the test surface or heat exchanger and apply energy balances to it to yield the governing differential/integral equations. The derivation of these equations is based on simplifying assumptions which may or may not be valid during tests. It has been shown in the analysis of the single blow and cyclic techniques that axial conduction in the surface material is of considerable importance, yet in the other technqiues the effect of axial conduction in the metal walls of the heat exchanger has been neglected. The influence of axial conduction in the metal may be less marked when the fluid is a liquid whose specific heat is much greater than that of air. Some workers on the dynamic behavior of heat exchangers (3) & (45) have found that it may be necessary to include the effect of axial dispersion in the fluid; this is the combined effects of conduction, mixing and diffusion. Axial dispersion has been shown to be particularly important at low fluid velocities which corresponds to a high residence time in the heat exchanger. None of the test methods reviewed take this mechanism into account.

Heat transfer coefficients determined using transient techniques are strictly speaking dynamic in nature, however, most workers have used heat transfer coefficients from steady state correlations to predict the dynamic response of heat exchangers, although few have included the variation in heat transfer coefficient with temperature in their models. Of those that have, Privott & Ferrell (42) and Gilles (18) show that a better prediction of the transient response is obtained when this variation is included. In none of the testing techniques reviewed is the dependance of the heat transfer coefficient on temperature included and, therefore, the methods assume that an average heat transfer coefficient will adequately describe the heat transfer process over the range of the temperature change. An improvement in the prediction of the heat transfer coefficients could be reasonably expected should the mathematical analysis of the testing techniques be modified to include the temperature dependance of the heat transfer coefficient. However, this aspect is not pursued further in this report for the reasons given below.

Some of the disadvantages of building a mathematical model around a typical segment of a heat exchanger are: the uncertainties in the model resulting from the assumptions made, the mathematical complexity of the solution and that the model is often restricted to heat exchangers of the same type. Therefore, bearing in mind these and other factors discussed above, none of the test techniques reviewed can be applied without modification to industrial heat exchangers manufactured by Serck. Even with modification the disadvantages with the element building approach make it worthwhile considering an alternative. The proposed approach to testing heat exchangers which this report starts to investigate is presented in the following chapter.

NOTE TO SUPPLEMENT EQUATION 1.14, PAGE 27

The equation for σ^2 was taken from reference 44. Obviously, this equation should be dimensionless, however, it has the units kg/m. Unfortunately, a copy of the original work (4) could not be obtained. As it was not intended to use the equation, its correct form has not been pursued further.

CHAPTER TWO

BASIS, THEORY AND COMPUTER PROGRAMME DEVELOPMENT FOR THE NEW DYNAMIC METHOD

2.1. INTRODUCTION

An alternative approach to the problem of using the transient response of a heat exchanger to obtain heat transfer information would be to treat the heat exchanger as a "black box" with a series of inputs and outputs. Using a process identification technique the relationships between each input and output variable would be determined and a mathematical model developed which would adequately describe the dynamic behaviour of the heat exchanger. The model could then be used in conjunction with the initial part of the transient response curve for each input and output variable to predict the operation of the heat exchanger under steady state conditions, thereby giving a potentially significant reduction in testing time since the need to obtain stabilized conditions before measurement of the test parameters would be eliminated.

This approach to testing differs from the test techniques reviewed in the previous chapter in so far as the mathematical model would be based on terminal considerations - the system inputs and outputs, whereas those reviewed are structural because they are based on a differential element. In addition, the alternative approach would predict the steady state temperatures which are then analysed by conventional methods, whereas previous test techniques obtain the partial heat transfer coefficient(s) by comparing the response curve with the solution to the appropriate mathematical model.

This chapter introduces the method chosen to identify the dynamic response of the heat exchanger. However, time was not available to process the large amount of experimental data collected and required to develop a model and compare the subsequent steady state predictions with the results from the conventional steady state tests.

2.2. THEORY

There is a wealth of information in the literature on process identification techniques ranging from simple manual procedures to complex computational routines for online identification required for adaptive control schemes.

The experimental work to be described in Chapter Four has been designed to assess the principal of the dynamic testing concept outlined above. By restricting the investigation to applying a single step input temperature disturbance, the response of the test heat exchanger can be expected to be a well ordered process, essentially noise free. Thus, the application of stochastic or statistical techniques is not appropriate to this immediate investigation and a simpler approach can be adopted.

It is well known that when dealing with the response characteristics of processes it is more convenient to manipulate data in the frequency domain rather than in the time domain. This requires the conversion of data from the time domain to the frequency domain and vice versa. This operation can be performed using the Fourier Integral which is frequently expressed by the Fourier Transform pair:

$$f(t) = \frac{1}{2\pi} \int_{-\infty}^{\infty} G(\omega) e^{j\omega t} d\omega = F^{-1}\left\{G(\omega)\right\} \tag{2.1}$$

$$G(\omega) = \int_{-\infty}^{\infty} f(t) e^{-j\omega t} dt = F\left\{f(t)\right\} \tag{2.2}$$

where $F^{-1}\left\{G(\omega)\right\}$ = the Inverse Fourier Transform, which enables the time function f(t) to be obtained when the frequency behaviour $G(\omega)$ is known.

$F\left\{f(t)\right\}$ = the Direct Fourier Transform, which enables the frequency response $G(\omega)$ to be determined from the time function f(t).

The numerical procedure for the Direct Fourier Transform is developed in Appendix A2.1. By arranging that the final value of f(t) is zero, the procedure collapses to the following summation:

$$G(\omega) = \frac{-jE_o}{\omega} - \frac{1}{\omega^2} \sum_{n=o}^{k-1} \frac{(E_{n+1}-E_n)}{\Delta t} \left[(\cos \omega t_n - \cos \omega t_{n+1}) - j(\sin \omega t_n - \sin \omega t_{n+1}) \right] \quad (2.3)$$

provided $f(t_k) = 0$

where ω_i = ith frequency

$E_n = f(t_n)$

$E_{n+1} = f(t_{n+1})$

t_n = time at sample n

t_{n+1} = time at sample n+1

Δt = sample interval

k = total number of f(t) samples

The amplitude of $G(\omega)$ is given by:

$$A(\omega_i) = \sqrt{(\Sigma \mathrm{Re})^2 + (\Sigma \mathrm{Im})^2} \quad (2.4)$$

and the phase angle by:

$$\phi(\omega) = \tan^{-1} \frac{\Sigma \mathrm{Im}}{\Sigma \mathrm{Re}} \quad (2.5)$$

where $\Sigma \mathrm{Re}$ = sum of the real parts of equation 2.3

$\Sigma \mathrm{Im}$ = sum of the imaginary parts of equation 2.3

In order to use this procedure for $F\{f(t)\}$ it is necessary to arrange the time domain data in a suitable form. This is achieved by subtracting the value of f(t) at each sample interval from the final value, thereby satisfying the condition $f(t_{k-1}) = 0$. Making this substitution:

$$E_n = A_{k-1} - A_n \qquad n = 0,1,2 \ldots\ldots k-1 \tag{2.6}$$

where A_n = value of time domain function at time t_n.

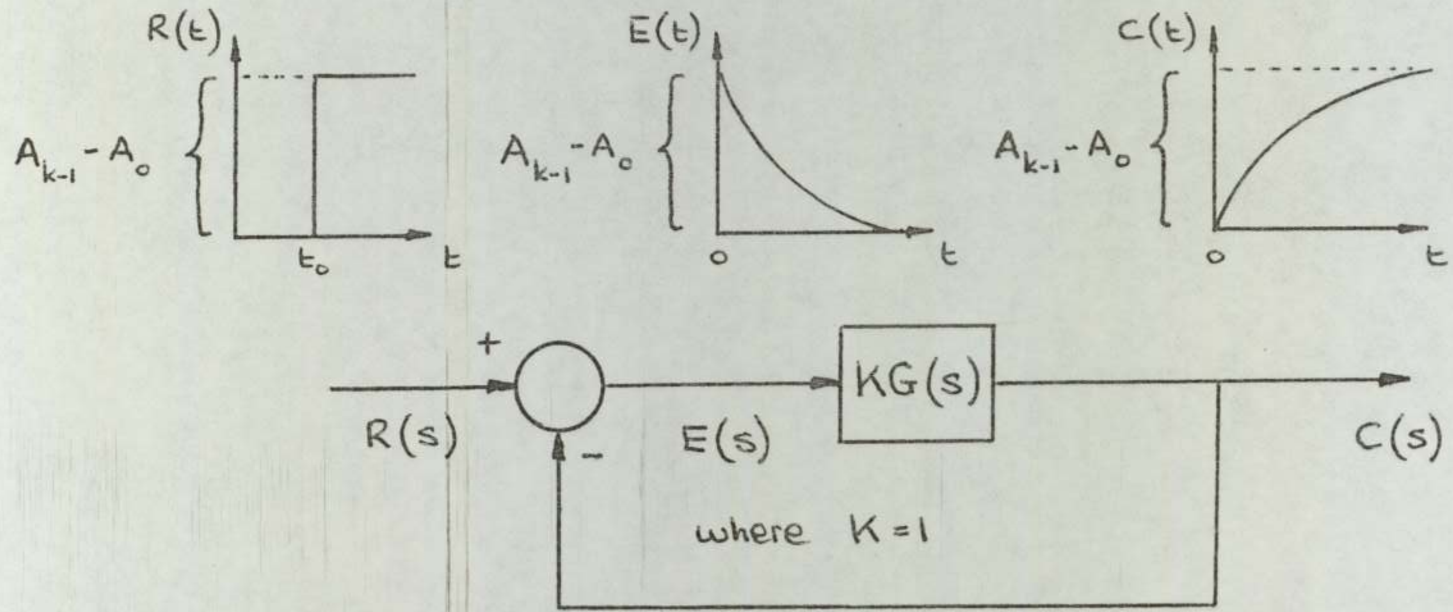
The data set represented by the E_i's is equivalent to the time domain error signal for a unity feed back circuit with unit gain subjected to a step input equal to $A_{k-1} - A_o$, fig. 2.1. Hence, the frequency response evaluated using equation 2.3 is for a fictitious error signal. This fictitious error signal arose because of the substitution made using equation 2.6 to simplify the calculation of $G(\omega)$ and will only become a true error signal when the input signal is a step input. Hereafter the fictitious error signal is referred to as the error signal unless otherwise stated.

The error signal frequency response for a linear dynamic system can be expressed as a ratio of two complex polynomials:

$$G(j\omega) = \frac{a_o + a_1(j\omega) + a_2(j\omega)^2 + \ldots\ldots + a_m(j\omega)^m}{1 + b_1(j\omega) + b_2(j\omega)^2 + \ldots\ldots + a_n(j\omega)^n} \tag{2.7}$$

using the algorithm of Sanathanan & Koerner (46). The roots of the two polynomials, and hence the poles, zeros and gain of $G(j\omega)$ can be found and yields the Laplace Transform of the error signal expressed in general form as:

$$E(s) = \frac{K \prod_{i=1}^{m} (s + z_i)}{\prod_{j=1}^{n} (s + p_j)} \tag{2.8}$$



$$E(s) = R(s) - C(s)$$

$$\therefore \quad C(s) = R(s) - E(s)$$

FIG. 2.1 BLOCK DIAGRAM REPRESENTATION OF EQUATIONS 2.6 & 2.9

$$\text{where } K = \frac{a_m}{b_n}$$

z_i = i th zero and p_j = j th pole

It is now necessary to convert the error signal E(s) to the true representation of the response signal in the s domain C(s). This is accomplished using the relationship:

$$C(s) = R(s) - E(s) \qquad (2.9)$$

$$\text{where } R(s) = \frac{A_{k-1} - A_o}{s}$$

for the unity feed back system, with unit gain, given in fig. 2.1. Details of the computational procedures referred to above are given in the next section.

2.3. DEVELOPMENT OF COMPUTER PROGRAMMES

To process the raw experimental data through to the stage where the input and output responses were expressed in Laplace Transform Form, four computer programmes were developed. The division of the data processing as represented by each programme is:

1. Programme SORT -Sorting experimental raw data into a suitable format for processing
2. Programme DFTR -Evaluation of frequency response
3. Programme FITFRQ-Curve fitting frequency response
4. Programme ESTOCS-Conversion of E(s) to C(s)

This suite of programmes was developed in order to handle conveniently the large quantity of experimental data and because it was anticipated that the oil/water case would be more complex than water/water heat exchange.

The major features of these programmes together with the programme flow charts are presented below. Programme listi-ings are given in Appendix A2.2. Fig 2,2 is a flow chart showing the general processing of the test data.

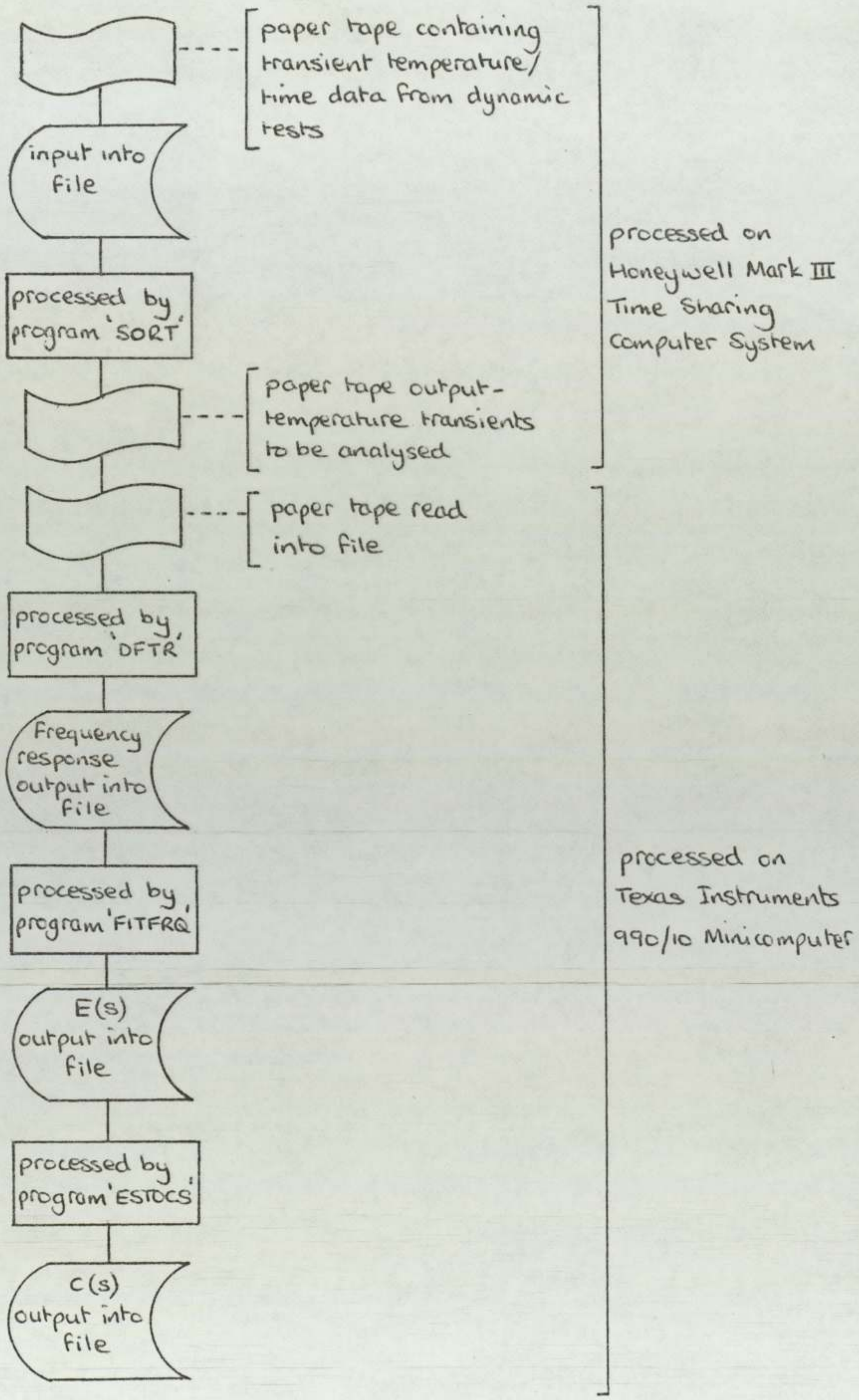


FIG. 2.2 FLOW CHART SHOWING GENERAL PROCESSING OF TEST DATA.

2.3.1. PROGRAMME 'SORT'

Programme SORT is used interactively to search for the start and finish of the transient response data; to separate this data from the total collected and to output the data for subsequent analysis.

The variation of the four inlet and outlet temperatures with time were recorded via a data logging system (described in Chapter 4) onto paper tape. One scan of the 4 thermopile voltages comprised 4 bursts of data per line as follows:

1st burst	- Shell inlet temperature data	
2nd "	- Shell outlet "	"
3rd "	- Tube inlet "	"
4th "	- Tube outlet "	"

Each burst of data consisted of 13 Asci II characters:

Characters	1-3	data logger scanner channel identification
	4	signal polarity
	5-9	digits in digital volt meter reading
	10-11	code to determine position of decimal point
	12-13	blank spaces

The programme converts the thermopile e.m.f.s to temperature using the calibration equations in Appendix A4.1.

Because the temperature measurements are not continuous but sampled, the accurate location of the start of the transient response is a problem.

Temperature versus time plots showed it was obvious that the transient response had started but not precisely when it commenced. To simplify the problem the start of the transient is assumed to occur just prior to where the 'obvious' change is first indicated. Inherent in this assumption is that any fluctuations in temperature occurring before the 'obvious' change are neglected. If present, these temperature variations would result from the combined effects of the test system response to the imposed step and any external disturbances, and to slight variations caused by incomplete fluid mixing.

The vicinity of the start and finish of each temperature transient was found by comparing the difference between successive temperature samples with a temperature difference parameter, X. If the condition:

$$|T(I) - T(I-1)| \text{ and } |T(I+1) - T(I)| \geqslant X \qquad (2.10)$$

where T(I) = i th temperature sample, is satisfied, then the start or finish of the temperature transient occurs between samples I-1 and I. The parameter X can be changed at the computer terminal to ensure correct location of the transient.

In the early stages of the programme development, a backward linear extrapolation method using the first two points on the response curve was used to estimate the starting point of the transient response. However, as can be seen from fig. 2.3 inaccurate location can result, as indeed was the case when some actual test data was processed.

To overcome this difficulty, it was assumed that the start of the transient occurred at the sample point immediately before the sample at which the response was first indicated, i.e. at sample I-1. For subsequent evaluation of the frequency response a linear relationship between each sample point in the transient response would be assumed. This approximation of the transient over the first sample interval was validated by visually inspecting the plot of the initial temperature responses at the four extreme conditions in flow and superimposing the above approximation.

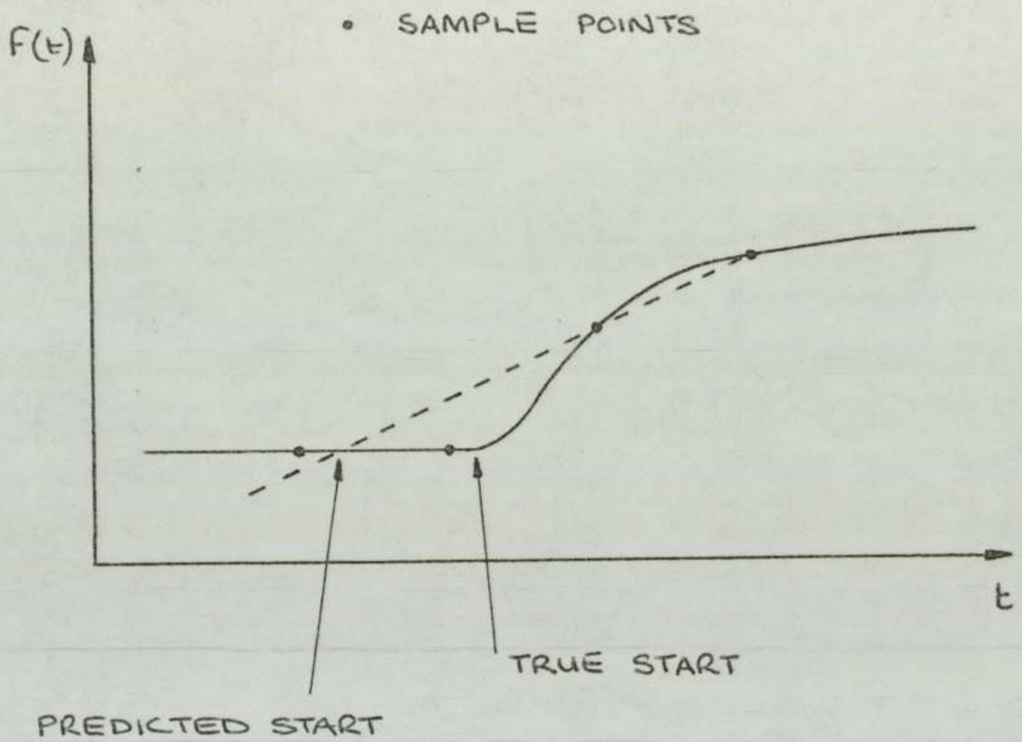
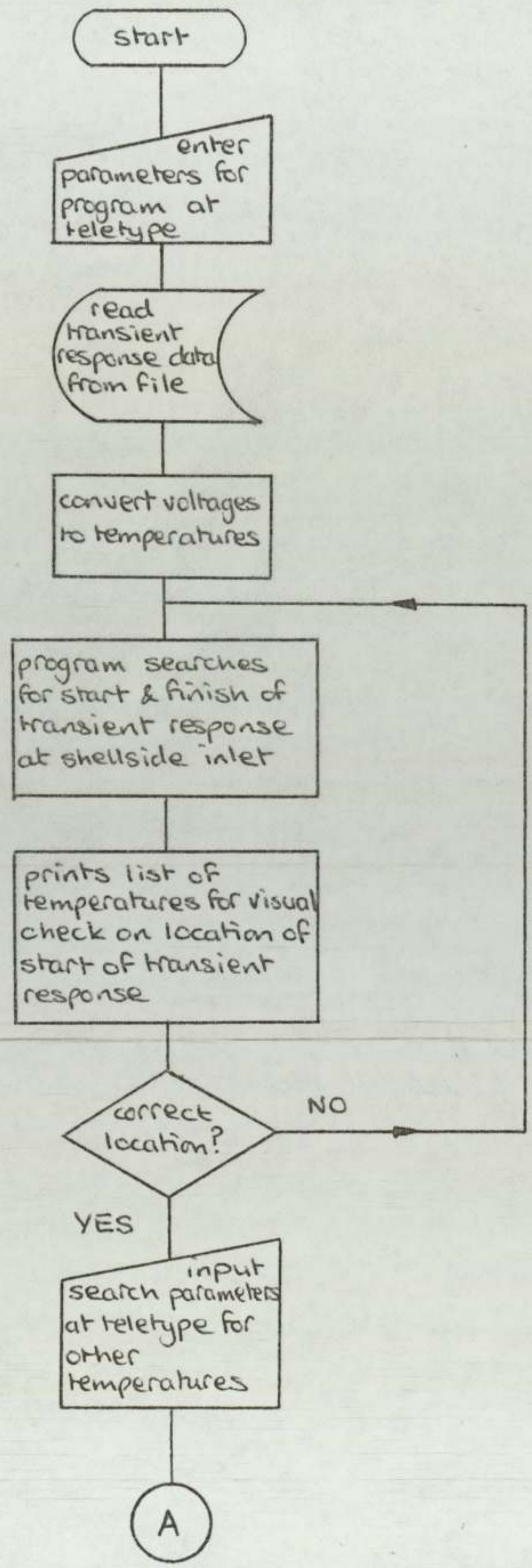
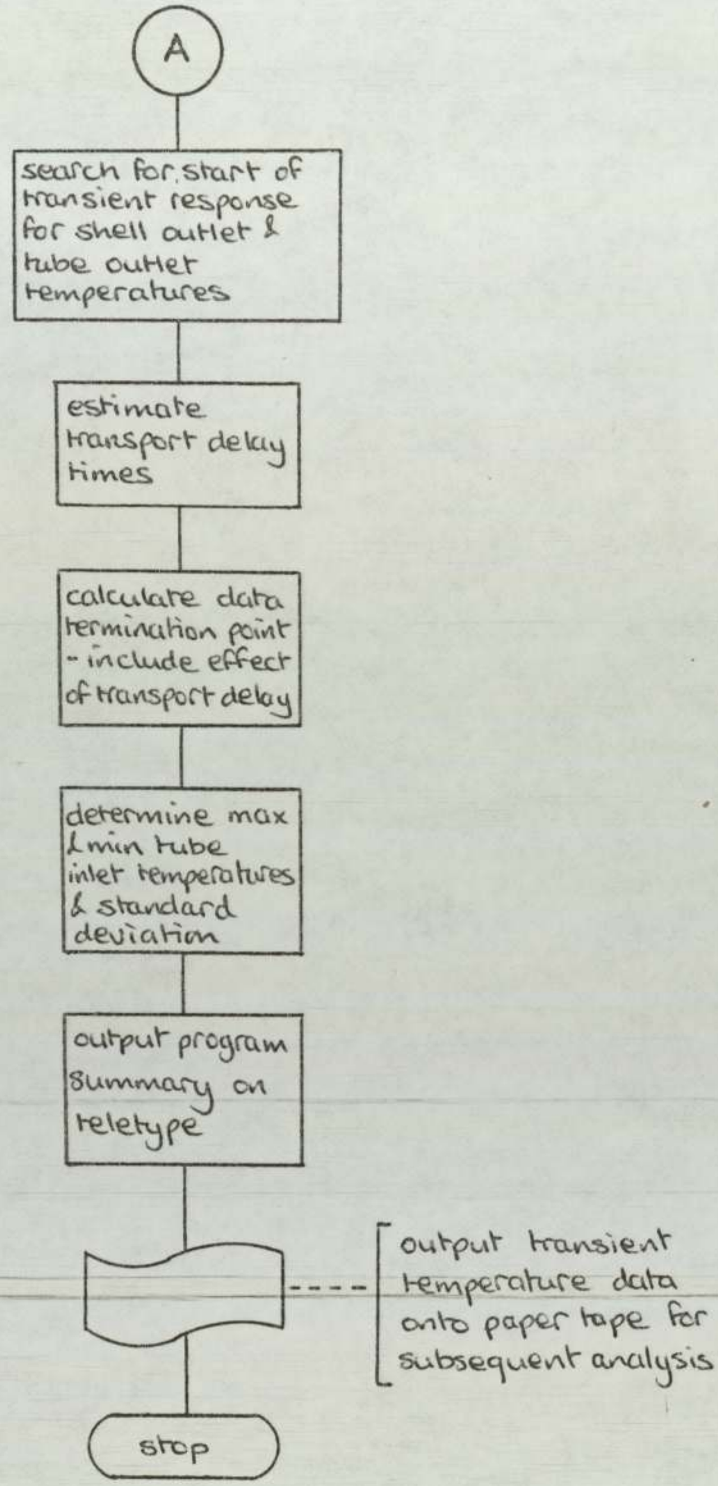


FIG. 2.3 INACCURACY OF BACKWARD LINEAR EXTRAPOLATION TO ESTIMATE START OF TRANSIENT RESPONSE.

FIG. 2.4 PROGRAMME 'SORT' FLOW CHART




A
search for start of transient response for shell outlet & tube outlet temperatures
estimate transport delay times
calculate data termination point - include effect of transport delay
determine max & min tube inlet temperatures & standard deviation
output program summary on teletype
output transient temperature data onto paper tape for subsequent analysis
stop

In test runs where a negative step input in temperature was applied following the stabilization of the positive step transient, the transport lag between the negative step being applied and it reaching the shell inlet was estimated and used to determine the final data point for the positive step transient. This was to ensure that any initial transient effects introduced by the negative step change were not included in the analysis of the positive step transient.

The transport delay times for the outlet temperature responses relative to the start of the response of the shell inlet were also estimated.

Throughout the tests it was desired to maintain the tube inlet temperature constant. To give a measure as to how well this was achieved the programme determined the maximum and minimum temperatures at the tube inlet for the duration of the transient and the corresponding standard deviation.

A flow chart for programme SORT is shown in fig. 2.4.

2.3.2. PROGRAMME 'DFTR'

Programme DFTR takes the transient response data from programme SORT and calculates the time domain error signal using equation 2.6. The frequency response is evaluated using a slight modification of the procedure given as equation 2.3. This equation is computationally more efficient when expressed as:

$$G(\omega) = \frac{-jE_o}{\omega} + \frac{(E_1 - E_o)}{-\Delta t\,\omega^2} + \sum_{n=1}^{k-1} (E_{n+1} - 2E_n + E_{n-1}) \left\{ \frac{\cos(-\omega t_n) + j\sin(-\omega t_n)}{-\Delta t\,\omega^2} \right\} \qquad (2.11)$$

The programme gives the flexibility to change the frequency increment over the frequency range of interest.

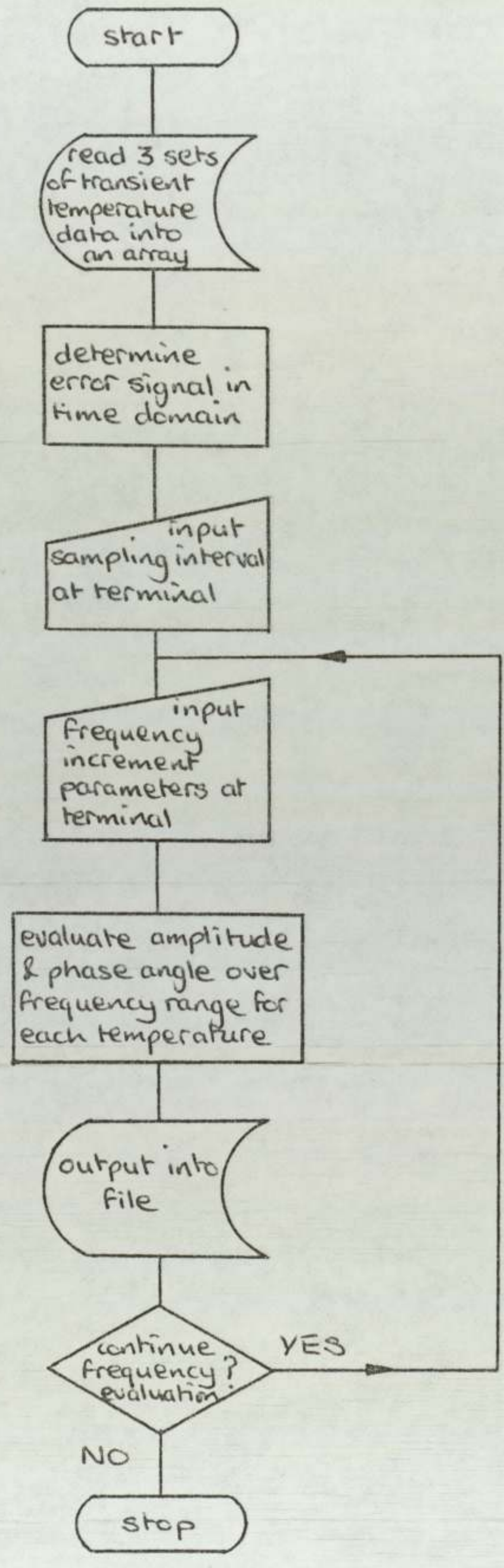
To evaluate the numerical accuracy of the routine, the function $e^{-t/T}$, where t is time and T is the time constant, was chosen, for which exact values for the frequency response could easily be calculated. The sample interval in the numerical routine was chosen as 1.2s and the sampling period as 180s. These values corresponded respectively, to the sampling rate and the period over which data was collected in the water/water tests. The frequency response was evaluated over the range $0.005 \leq \omega \leq 10$ rad/s for two time constants and is compared subjectively with the theoretical solutions below.

TIME CONSTANT(s)	NUMERICAL ACCURACY v THEORY
10	excellent
2.5	good

As the time constant reduces from 2.5s towards the sampling time of 1.2s and lower, the frequency response evaluation will clearly deteriorate because there will be insufficient samples to adequately describe the transient time response using a linear approximation between samples. Note, that for a time constant of 1.2s it is possible for 63% of the transient response amplitude to have occurred before a second sample has been taken. Therefore, in cases where the time constant of a simple exponential decay is <2.5s the accuracy of the numerical routine deteriorates. It is also reasonable to expect the procedure to break down for more complex functions, however, time did not permit a deeper study into the numerical accuracy of the Direct Fourier Transform routine.

Fig 2.5 shows the flow chart for programme DFTR.

FIG. 2.5 PROGRAMME 'DFTR' FLOW CHART.



2.3.3. PROGRAMME 'FITFRQ'

The procedure of Sanathanan & Koerner used in programme FITFRQ to express the frequency response as a ratio of two complex polynomials is an iterative technique, which evaluates the polynomial coefficients by minimising the sum of the squares of the error between the actual function and the polynomial ratio.

The roots of the two polynomials are found by a routine based on the Newton-Raphson procedure for complex polynomials and used to express the Laplace Transform of the error signal given by equation 2.8.

Equation 2.8 can be expressed alternatively as a series of time constants in the numerator and denominator:

$$E(s) = K' \frac{\prod_{i=1}^{m} (1 + T_i s)}{\prod_{j=1}^{n} (1 + T_j s)} \qquad (2.12)$$

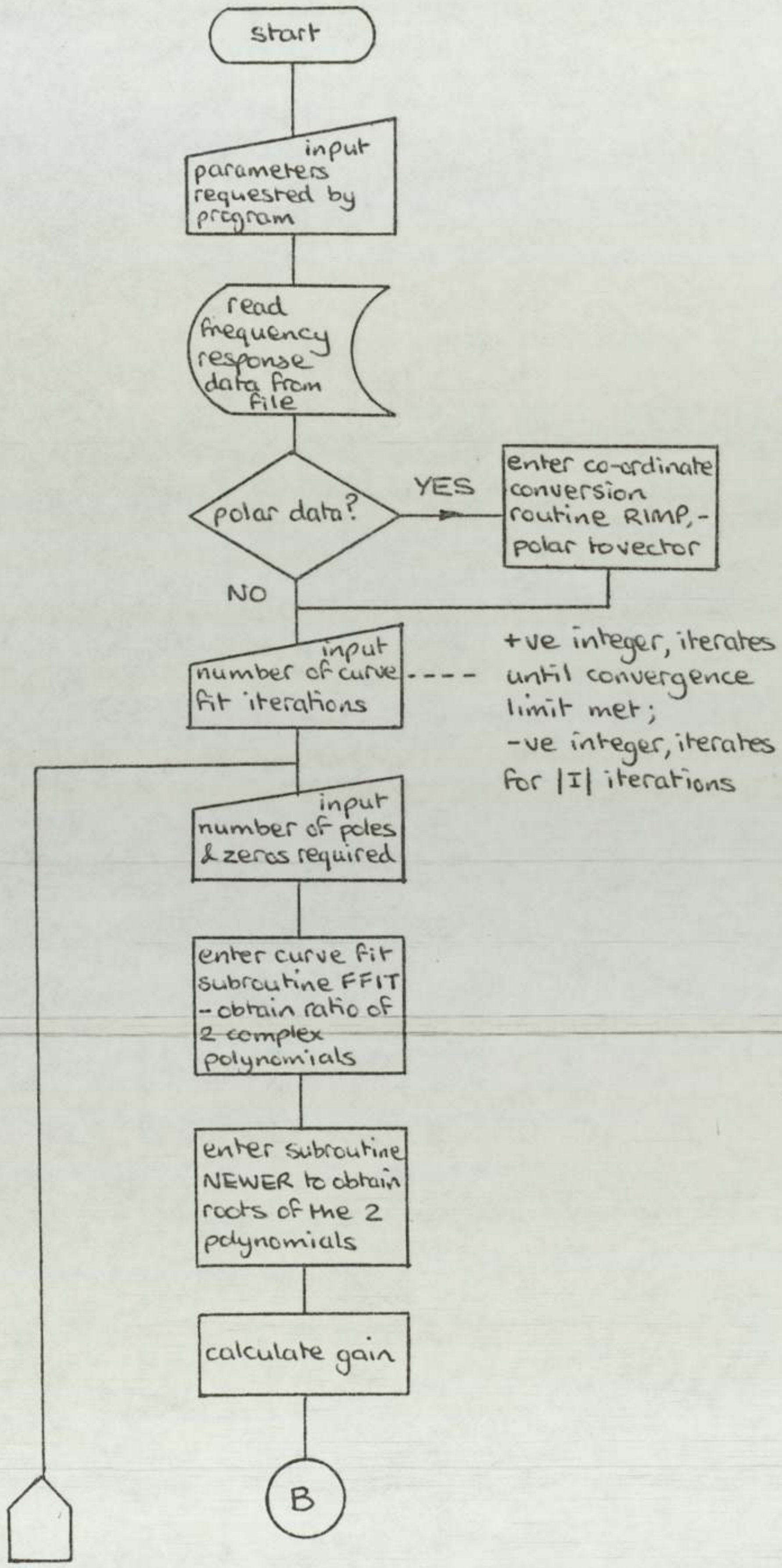
Where $T_i = \dfrac{1}{z_i}$

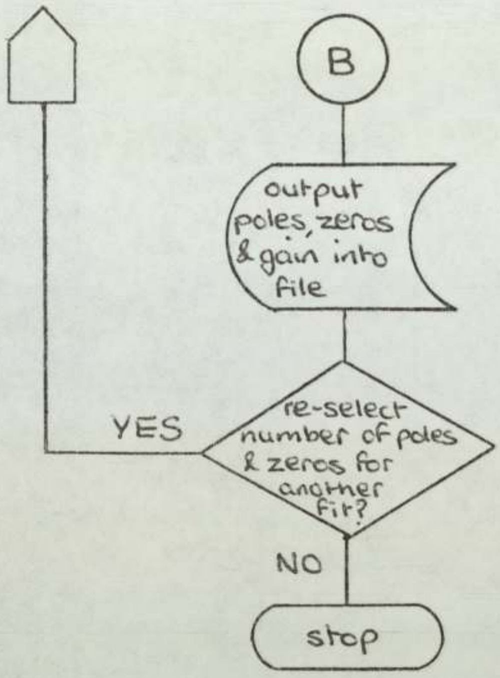
$$T_j = \frac{1}{p_j}$$

$$K' = K \frac{\prod_{i=1}^{m} z_i}{\prod_{j=1}^{m} p_j}$$

The curve fitting routine, FITFRQ, was originally developed by Madden (33), however, the master programme has been altered extensively to meet the requirements of this work and the subroutine GAIN2 added to calculate the gain corresponding to the time constants. A flow chart for programme FITFRQ is given in fig. 2.6, while in Appendix A2.2.3. the master programme FITFRQ and the subroutine GAIN2

FIG. 2.6 PROGRAMME 'FITFRQ' FLOW CHART




B
output poles, zeros & gain into file
re-select number of poles & zeros for another fit?
YES
NO
stop

are given as programme listings.

2.3.4. PROGRAMME 'ESTOCS'

Programme ESTOCS converts the error signal response (equation 2.8) to the true representation of the input and output responses. The programme is based on equation 2.9 which when the appropriate substitutions for R(s) and C(s) are made gives:

$$C(s) = \frac{K_o}{s} - \frac{K\prod_{i=1}^{m}(s + z_i)}{\prod_{j=1}^{n}(s + p_j)} \qquad (2.13)$$

where $K_o = A_{k-1} - A_o$ = final temperature - initial temperature.

ESTOCS consists of a series of subroutines to manipulate equation 2.13 into a similar form to that expressing E(s) in equation 2.8, thus:

$$C(s) = \frac{K_o\prod_{j=1}^{n}(s + p_j) - Ks\prod_{i=1}^{m}(s + z_i)}{s\prod_{j=1}^{n}(s + p_j)} \qquad (2.14)$$

$$= \frac{K_1\prod_{u=1}^{q}(s + z_u)}{\prod_{v=1}^{n+1}(s + p_v)} \qquad (2.15)$$

$$\text{where } q = \begin{cases} n, & \text{for } n \geqslant m+1 \\ m+1, & \text{for } n < m+1 \end{cases}$$

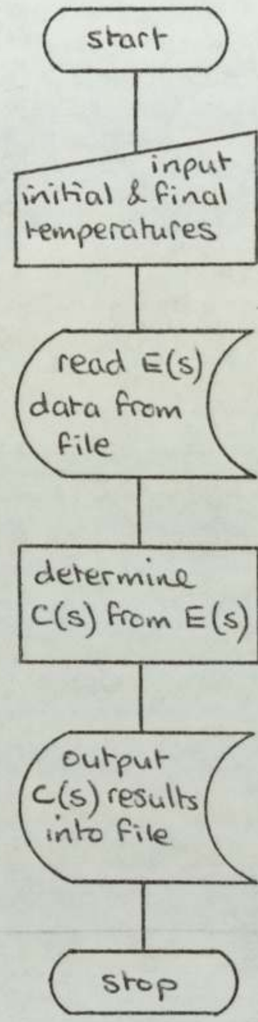
$$K_1 = \frac{\text{coefficient of } s^q}{\text{coefficient of } s^{n+1}}$$

In a similar manner to E(s), C(s) can be expressed in terms of time constants.

The flow chart for programme ESTOCS is shown in fig. 2.7.

The set of four programmes is used in the analysis to be described in Chapter 5.

FIG. 2.7 PROGRAMME 'ESTOCS' FLOW CHART.



CHAPTER THREE

EXPERIMENTAL WORK PART ONE - STEADY STATE TESTS

The purpose of the steady state heat transfer performance tests was to establish for the test heat exchanger the datum performance curves against which the results predicted from the dynamic tests would be evaluated to determine the suitability or otherwise of the dynamic testing method. For this evaluation the overall heat transfer coefficient, which is a direct measure of the heat exchanger's thermal performance, was to be used. Since the dynamic technique predicts the steady state temperatures it is not necessary to separate the partial heat transfer coefficients from the overall coefficient as part of the evaluation. This is because provided the overall coefficient derived from steady state tests is equal to the overall coefficient predicted by the dynamic test method, the derived partial coefficients will be identical.

3.1. THE TEST HEAT EXCHANGER

The heat exchanger used throughout the investigation was a standard Serck shell and tube type production unit, designated AA39-B KEF7. It is a small unit with one tube-side fluid pass and one shellside fluid pass with disc and doughnut baffle arrangement and tested in counterflow. The test unit has an overall length of 279mm, a cylinder outside diameter of 86mm and is manufactured to Serck general arrangement drawing 45511-4101. The heat exchanger comprises an aluminium brass tubestack encased in an aluminium cylinder with gun metal water boxes. A complete description of the unit, plus photograph, is given in Appendix A1.

3.2. THE HEAT TRANSFER TEST RIGS

The performance tests were carried out on standard heat transfer test rigs in the company's Engineering Laboratory.

For the water/water tests the hot water was passed over the tubes (on the shellside)and the cold water through the tubes (the tubeside), while for the oil/water tests the hot oil was on the shellside of the unit and the water on the tubeside. The circuit diagrams for the water/water and oil/water test rigs are shown in figures 3.1 & 3.2 respectively.

Fluid temperatures were measured with Class A, BSI or NPL calibrated, mercury in glass thermometers positioned in thermometer pockets located near the test heat exchanger's inlet and outlet ports. To ensure that the fluid was well mixed at the point of temperature measurement the pipe diameter was chosen to give a high Reynolds Number to promote good fluid mixing and/or mixing orifices inserted. The test unit, thermometer pockets and connecting pipework were well lagged to minimise heat losses.

Water mass flow rates were measured using orifice plates in orifice units with D and D/2 pressure tappings manufactured to BS1042 Part 1:1964. The calibration curve was generated from the British Standard and verified by dead-weight measurement. The pressure differentials across the orifice plates were measured with mercury 'U' tube manometers. The oil flow rate was determined by timing the flow of a fixed volume of oil as indicated by a totalising type flow meter. This flow meter had been previously calibrated by dead-weight measurement.

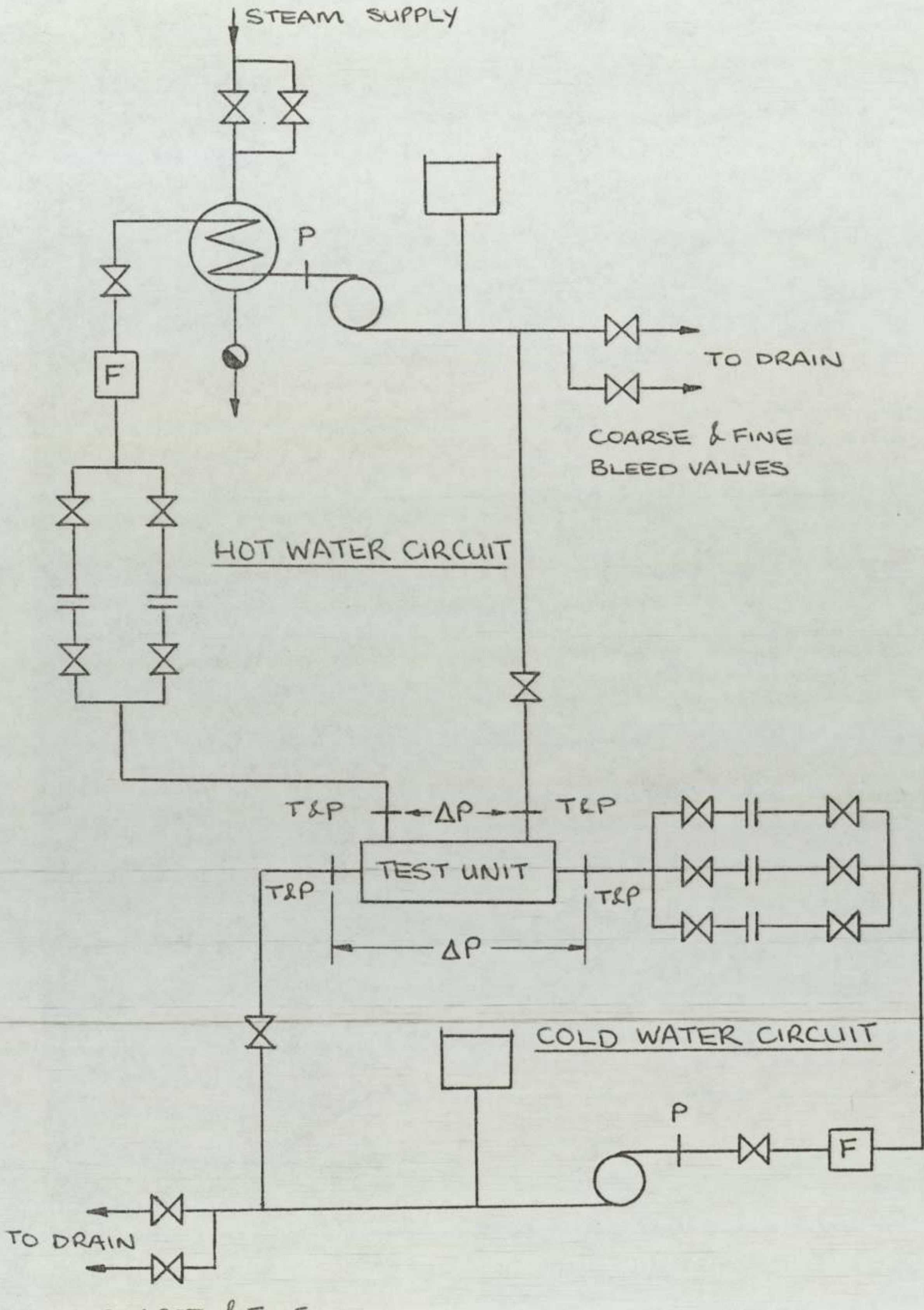


FIG. 3.1 WATER/WATER STEADY STATE TEST RIG CIRCUIT DIAGRAM (KEY TO SYMBOLS BELOW FIG. 3.2)

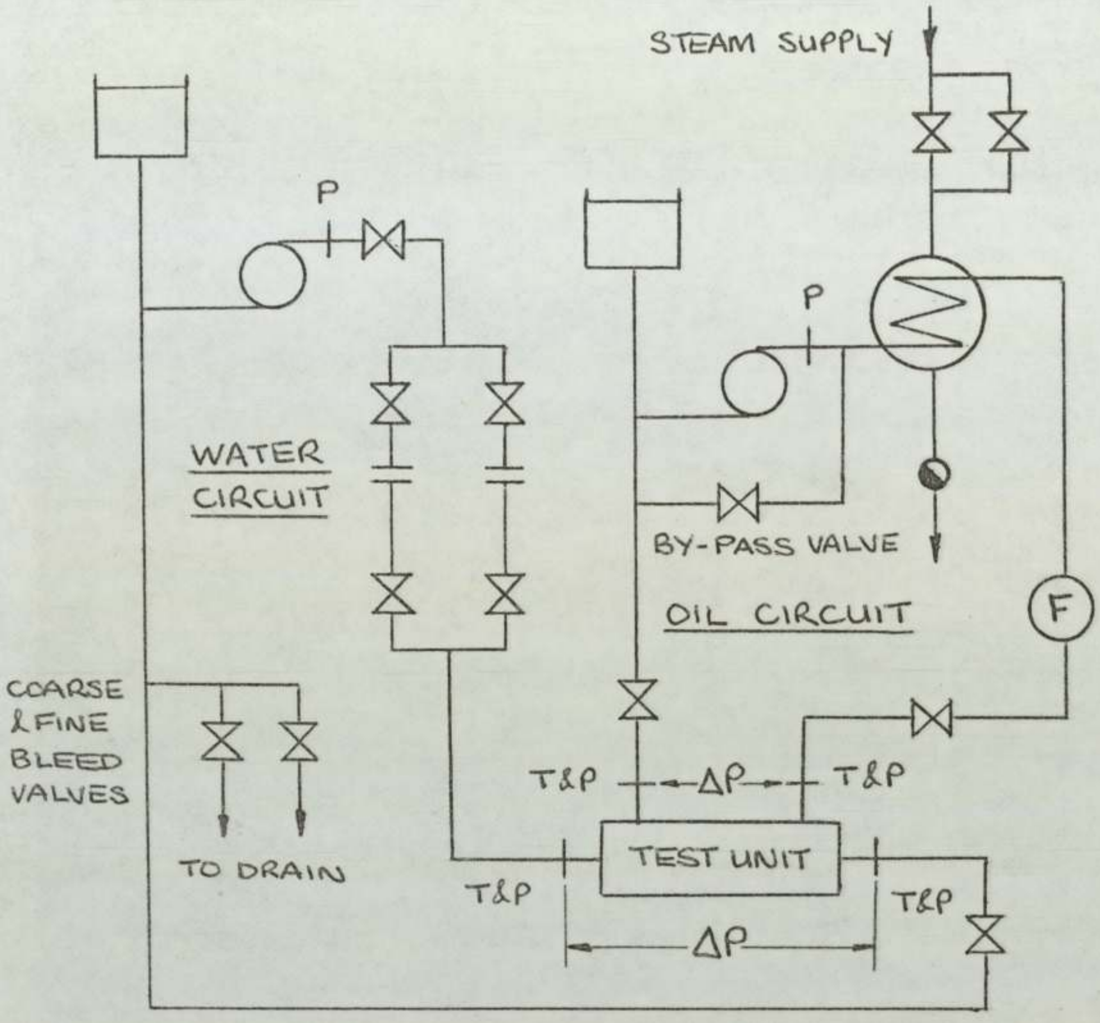
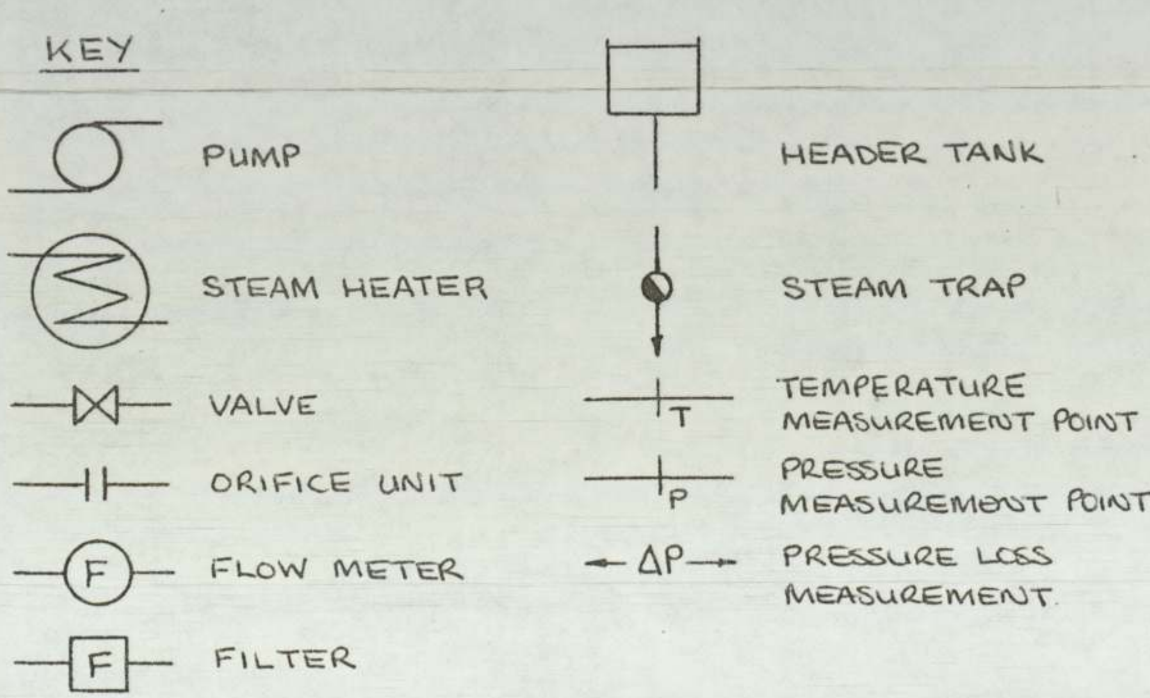


FIG. 3.2 OIL/WATER STEADY STATE TEST RIG CIRCUIT DIAGRAM



3.3. TEST PROCEDURE

After initial start up of the test rig all the air bleed valves are opened and the pipework, manometer and pressure leads purged of any air present.

The water flow rate is changed by throttling the valve immediately downstream of the heat exchanger outlet. The oil flow is controlled by the valve downstream of the test unit outlet and the pump by-pass valve.

The two circuit diagrams for the test rigs, figs. 3.1 & 3.2, show that the hot fluid circuit is heated utilising steam. The cold fluid circuit is brought up to temperature by using the test unit as a heater. Temperature stabilization to the desired temperature is achieved by the combined adjustment of the steam flow through the steam heater and the water bleed valves. The latter allows hot water to be bled to waste and to be replaced by cold water from the header tank. Because of the interaction between the two fluid circuits at the test unit a change in conditions in one circuit will produce a change in the other, and, in addition, the flow rate will usually vary with change in temperature. Consequently it can take a considerable time before thermal equilibrium at the required test point conditions is achieved.

As has been previously discussed in the introduction to this report, Section I.3.2, it is essential to allow the heat transfer rig to stabilize thermally before heat transfer measurements are taken in order to obtain reliable test data. One method of ensuring that the test rig has reached a state of equilibrium is to take a set of temperature readings at inlet and outlet to the test unit at frequent intervals for fixed flow rates. When successive readings continue to show negligible change, the test rig can be considered as being stabilized and heat transfer data recorded. The inlet and outlet temperatures at the test unit are then recorded in sequence over a period of one minute. This procedure is repeated five times during which the temperature at any location must not change by more than 0.1K. If this condition is not satisfied the series of readings are rejected and the test rig allowed to further stabilize. The temperature readings indicated by the thermometers are then corrected by applying the thermometer calibration and stem corrections and the heat balance calculated. The heat balance gives the tester an initial indication of the reliability to be expected of the test data.

It is usual to maintain the tube flow rate constant and vary the shell flow over the desired range, then to change the tube flow and repeat the procedure for the remaining test conditions.

3.4. TEST CONTENT

In general, heat transfer thermal performance tests are carried out over a range of shell and tubeside flow rates which cover the full expected operational range of the heat exchanger. For convenience a series of shell flows at each of a number of tube flow rates are often chosen, although the total number of test points depends on the test requirement. For the particular test unit used in this study three heat transfer performance tests were carried out. The inlet temperatures to the test unit were arbitrarily set at 50°C for the hot shellside fluid and 20°C for the tubeside water. This was to give a reasonable temperature difference between the two fluids and ease the problems associated with maintaining a constant fluid supply temperature in the dynamic tests. The first performance test was for the water/water case and the test unit tubestack was degreased and bright dipped prior to testing to ensure the heat transfer surface was clean. A series of repeat points were chosen to demonstrate the repeatability of the testing and to assess whether or not the heat transfer surface had become fouled during the tests, (see section 3.6). Upon completion of this first test the aluminium cylinder of the test unit was found to be cracked and was replaced. Therefore, to ensure the correct datum was established, the test unit was retested. The third test was the oil/water performance test; the oil used was Shell Diala B. Prior to this test the tubestack was cleaned. A summary of the test conditions for the three performance tests is given in Table 3.1.

3.5. ANALYSIS OF TEST RESULTS

The test results were processed using computer programme HEAT for which a flow chart and listing is given in Appendix A3.2. A summary of its function is given below.

The programme calculates the overall heat transfer coefficient U for each combination of shell and tube flow rates and estimates the maximum uncertainty in U. From equation I.6 with $F_c = 1$:

$$U = \frac{Q}{A_o \Delta\theta_{LMTD}}$$

where the evaluation of the heat transfer rate Q is based on either the hotside or coldside dissipation (equation I.7 or I.8) depending for which calculation the estimated maximum uncertainty is the smaller. Details of the uncertainty analysis for the steady state test results are given in Appendix A3.1. The equation for the variation in the specific heat of water with temperature was determined using data values taken from the 1970 Steam Tables and the curve fitting routine POLFIT in the Honeywell STATSYST*** package. The equation for the specific heat of Shell Diala B oil was obtained from the correlating equation given in shell publication, "Liquid Phase Heat Transfer", 1972, page 9. The computer print out for each of the three performance tests is given in Appendix A.3.3. Specimen calculations are included as Appendix A3.4.

The results of each of the performance tests are presented as carpet plots of shell and tube flow rates versus the overall heat transfer coefficient in figs. 3.3 - 3.5. This method of presentation is a useful way of showing how changes in two independent variables (shell and tube flowrates) affect the dependant variable (the overall coefficient). It is particularly useful when actual test flow rates do not match exactly the test conditions required, then, utilising the carpet plot technique the overall coefficient corresponding to the required shell and tube flow rates can be reliably estimated by what amounts to graphical interpolation.

3.6. DISCUSSION OF TEST RESULTS

The results of the steady state tests presented in figs. 3.3 - 3.5 show that the data is consistent, with the overall heat transfer coefficient increasing with increase in both shell and tube fluid flow rates as would be expected and, additionally, with all the data points falling reasonably close to the intersection of the grid lines on the carpet plot.

Fig. 3.6 shows the results of the second water/water tests superimposed on those of the first. The discrepancy between the two sets of results varies between ± 10%. There is insufficient test data available to ascertain the true reasons for this discrepancy, although an increase in fluid by-passing on the shellside caused by the change in cylinder may be expected to produce a decrease in performance, (see section I.3.2, final paragraph). The second water/water test data is consistent with this premise except at the lowest tube flow rate.

It is usual to have some test points repeated, particularly when heat balances greater than 10% result and in order to assess whether or not the heat transfer surface has become fouled during the tests, which will yield unreliable data. Repeat points also demonstrate the repeatability of test data.

To assess if fouling had occurred during the first water/water tests, points taken near the end of these tests were compared with points recorded at similar conditions at the beginning and midway through the tests. With the exception of one repeat point (point 26) a slight drop in performance resulted. The percentage decrease in the value of the overall heat transfer coefficient was less than the estimated maximum uncertainty in the original calculation of the overall coefficient - see Table 3.2. It is shown in Appendix A3.5 that when a suitable allowance for fouling is made a significant reduction in performance occurs, $\gg$ 20%. Since the magnitude of the actual decrease in the overall coefficient is very much less than would be expected if the heat exchanger surface had become fouled it is concluded that negligible fouling of the test unit occurred during the tests. The repeat points also give an indication as to the repeatability of the test data, which for the repeat points selected was better that 4%.

A similar argument concerning the assessment of possible fouling applies to the second water/water tests and hence it is concluded that negligible fouling occurred. However, for these tests the percentage change in the overall coefficient for the repeat points was greater than the corresponding maximum uncertainty in the initial test points - Table 3.3. The three repeat test points lay within the range - 7.1% to + 9.5% of the original points.

For the oil/water tests it is not possible to make a fouling assessment because the repeat points were performed soon after the original test points. Appendix A3.5 again shows that a marked reduction in the overall coefficient can be expected if appreciable fouling takes place. However, since the oil used in the tests was clean and there were no visible signs of fouling present on the tubestack it is concluded that the performance of the test unit has not deteriorated due to fouling. The repeat points do give an indication of their repeatability which was better than 2.5% and all were within the maximum uncertainty limits - Table 3.4.

Heat balances for both water/water tests were generally better than 5% which from previous Company experience is satisfactory. For the oil/water tests the heat balances were generally better than 7%, although two test points showed heat balances in excess of 12% which is regarded as being unsatisfactory. However, this occurred when the tubeside temperature difference was 0.5K. If this temperature difference was in error by 0.05K a more acceptable heat balance would result. It was therefore decided to accept these two test points.

The estimated maximum uncertainties in the water/water and oil/water test results were better than $\pm$ 7% and $\pm$ 9% respectively. It should be noted that the uncertainty analysis does not include an uncertainty component accounting for the possibility of fouling of the heat exchanger surface.

The first water/water test results seem to be generally better results than those of the second test because the repeat points lie within the bounds of the initial uncertainties and in the second test they do not. However, they cannot be used as datum performance curves because a replacement cylinder was used in the second water/water tests and subsequent dynamic tests since upon completion of the first tests the original cylinder was found to be cracked. Therefore, the second water/water tests are cautiously adopted as the datum performance curves for the water/water case.

The oil/water results show excellent consistency and give confidence in their adoption as the reference performance curves for the oil/water case.

TABLE 3.1. TEST CONDITIONS FOR STEADY STATE HEAT TRANSFER TESTS

NOTE: The flow rates may not correspond exactly to the flow rates recorded on test.

Shellside fluid inlet temperature $50^{o}C \pm 1K$

Tubeside fluid inlet temperature $20^{o}C \pm 1K$

(a) FIRST WATER/WATER TESTS

		TUBESIDE WATER FLOW RATE (kg/s)				
		0.151	0.340	0.756	1.512	2.646
SHELLSIDE WATER FLOW RATE (kg/s)	0.151	X R	X	X	X	X R
	0.302	X		X	X	X
	0.529	X R		X	X	X R
	0.756	X R	X	X	X	X R

Shellside Re range: 3100 - 15500

Tubeside Re range: 580 - 10200

(b) SECOND WATER/WATER TESTS

		TUBESIDE WATER FLOW RATE (kg/s)				
		0.151	0.340	0.756	1.512	2.646
SHELLSIDE WATER FLOW RATE (kg/s)	0.151	X	X	X R	X R	X
	0.302	X	X	X	X	X
	0.529	X	X	X	X	X
	0.756	X	X	X R	X	X

Shell and tubeside Re range as above.

TABLE 3.1 CONTINUED

(c) OIL/WATER TESTS

		TUBESIDE WATER FLOW RATE (kg/s)			
		0.151	0.340	0.756	1.512
SHELLSIDE OIL FLOW RATE (kg/s)	0.151	X	X	X R	X R
	0.302	X	X	X	X
	0.529	X	X	X R	X
	0.756	X	X	X R	X R

Shellside Re range: 210 - 1050.

Tubeside Re range: 580 - 5800.

X - TEST CONDITION.

R - REPEAT TEST POINT.

TABLES 3.2-3.4 SHOW THE ESTIMATED MAXIMUM UNCERTAINTY IN THE OVERALL HEAT TRANSFER COEFFICIENT AND THE PERCENTAGE DIFFERENCE BETWEEN REPEAT TEST POINTS AND THE ORIGINAL TEST POINTS.

Note: The flow rates may not correspond exactly to the flow rates recorded on test.

TABLE 3.2. UNCERTAINTY IN THE FIRST WATER/WATER TESTS

SHELL FLOW RATE (kg/s)	TUBE FLOW RATE (kg/s)				
	0.151	0.340	0.756	1.512	2.646
0.151	1 & 23* 5.8 & 5.9%	18 5.4%	8 5.0%	9 4.9%	16 & 24 4.8%
0.302	2 5.8%	-	7 5.5%	10 5.1%	15 5.0%
0.529	3 & 22 5.7 & 5.6%	-	6 6.2%	11 5.5%	14 & 25 5.2%
0.756	4 & 21 5.6%	19 6.2%	5 6.8%	12 5.9%	13 & 26 5.5%

* test point number

Note: no test point number 17

ORIGINAL POINT NO.	REPEAT POINT NO.	% AGE DIFFERENCE
1	23	-1.8
3	22	-4.0
4	21	-3.2
16	24	-3.7
14	25	-1.7
13	26	+1.0

$$\% \text{ age difference} = \left[\frac{U_{\text{repeat}}}{U_{\text{original}}} - 1\right] \times 100$$

TABLE 3.3. UNCERTAINTY IN THE SECOND WATER/WATER TESTS

SHELL FLOW RATE (kg/s)	TUBE FLOW RATE (kg/s)				
	0.151	0.340	0.756	1.512	2.646
0.151	11 5.8%	5 5.4%	3 & 23 5.1%	14 & 21 4.9%	19 4.8%
0.302	12 5.8%	6 6.2%	4 5.5%	13 5.2%	20 5.1%
0.529	10 5.5%	7 6.4%	1 6.2%	15 5.6%	18 5.5%
0.756	9 5.5%	8 6.2%	2 & 22 6.6 & 6.7%	16 5.9%	17 5.0%

ORIGINAL POINT NO.	REPEAT POINT NO.	% AGE DIFFERENCE
3	23	+6.7
2	22	-7.1
14	21	+9.2

TABLE 3.4. UNCERTAINTY IN THE OIL/WATER TESTS

SHELL OIL FLOW RATE (kg/s)	TUBE FLOW RATE (kg/s)			
	0.151	0.340	0.756	1.512
0.151	1 7.4%	5 7.1%	9 & 13 7.0%	14 & 21 6.8%
0.302	2 6.9%	6 7.9%	10 7.5%	15 7.3%
0.529	3 6.5%	7 8.4%	11 & 18 8.1%	16 7.8%
0.756	4 6.2%	8 8.0%	12 & 19 8.7%	17 & 20 8.3 & 8.4%

ORIGINAL POINT NO.	REPEAT POINT NO.	% AGE DIFFERENCE
9	13	-1.4
11	18	-2.2
12	19	-1.5
14	21	0.0
17	20	-0.5

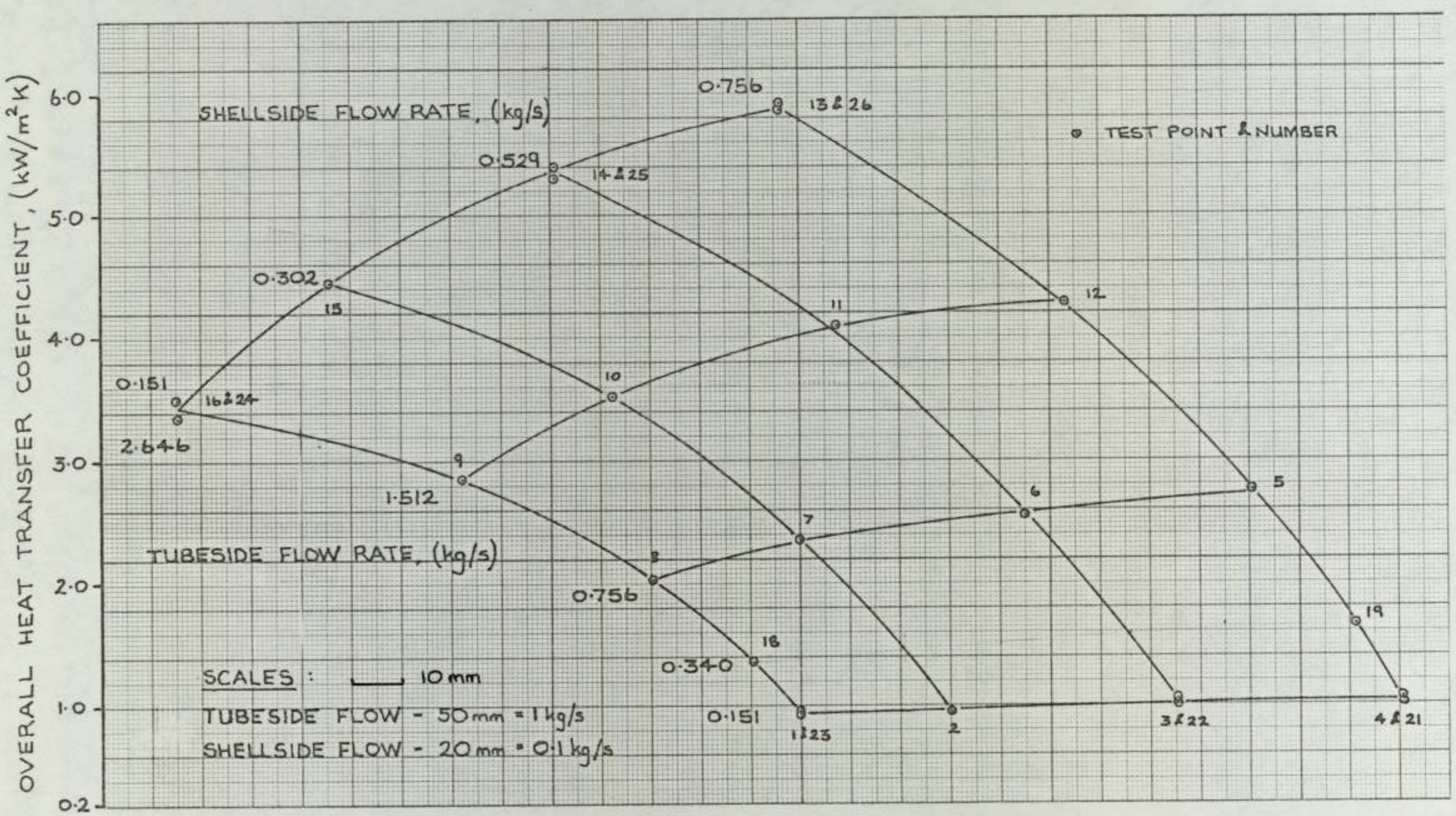


FIG. 3.3 SHELL AND TUBE FLOW RATES VERSUS OVERALL HEAT TRANSFER COEFFICIENT FOR THE FIRST WATER/WATER TESTS.

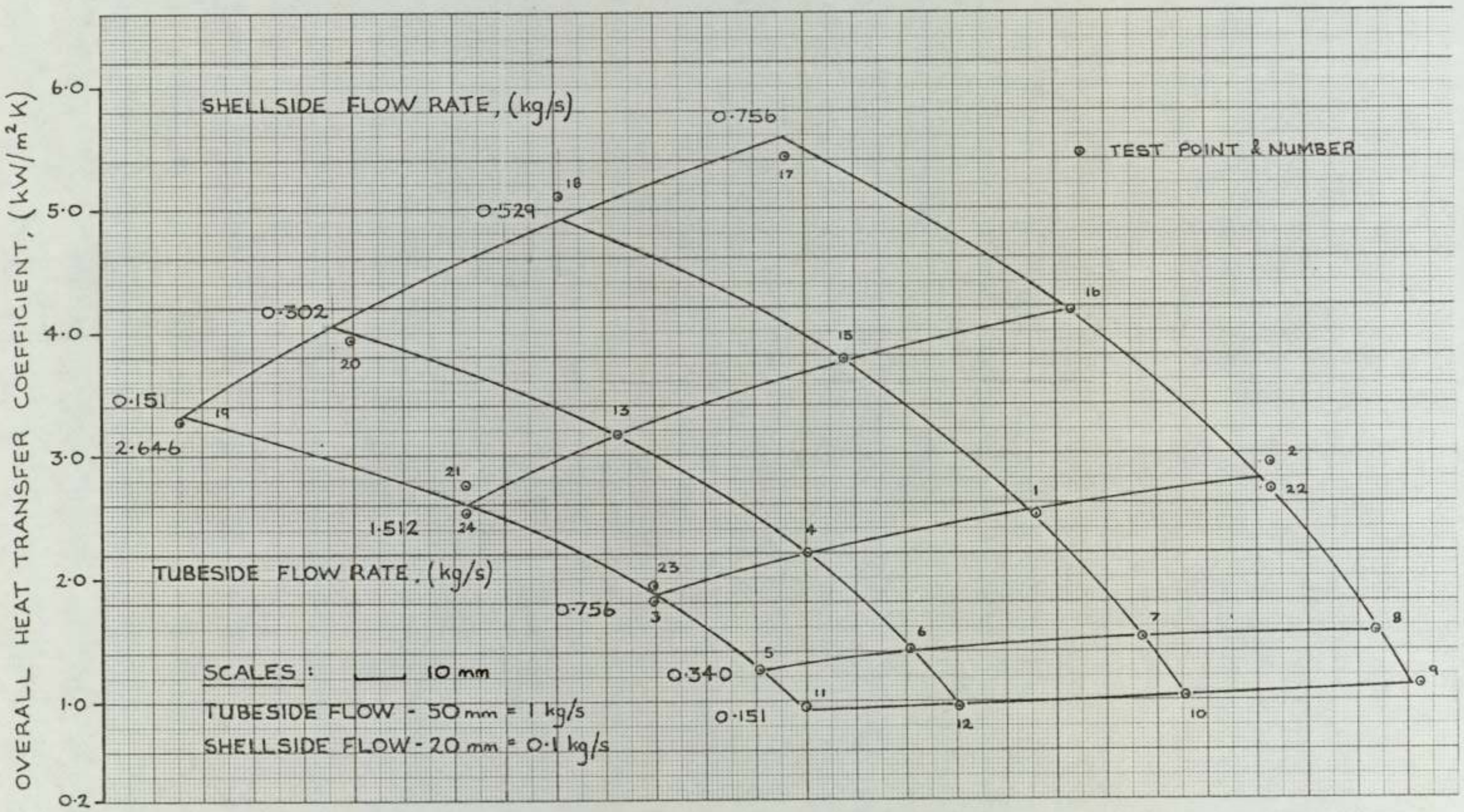


FIG. 3.4 SHELL AND TUBE FLOW RATES VERSUS OVERALL HEAT TRANSFER COEFFICIENT FOR THE SECOND WATER/WATER TESTS.

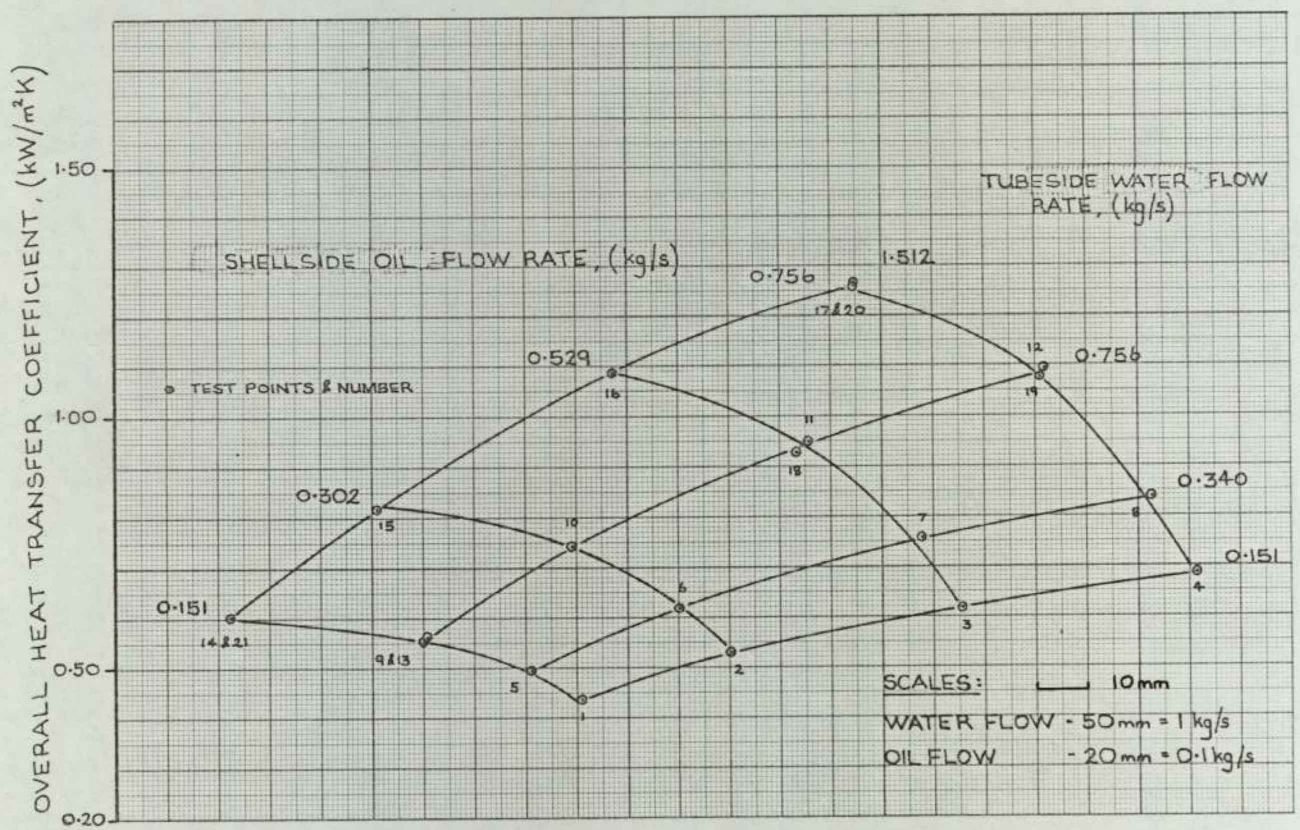


FIG. 3.5 SHELL AND TUBE FLOW RATES VERSUS OVERALL HEAT TRANSFER COEFFICIENT FOR THE OIL/WATER TESTS.

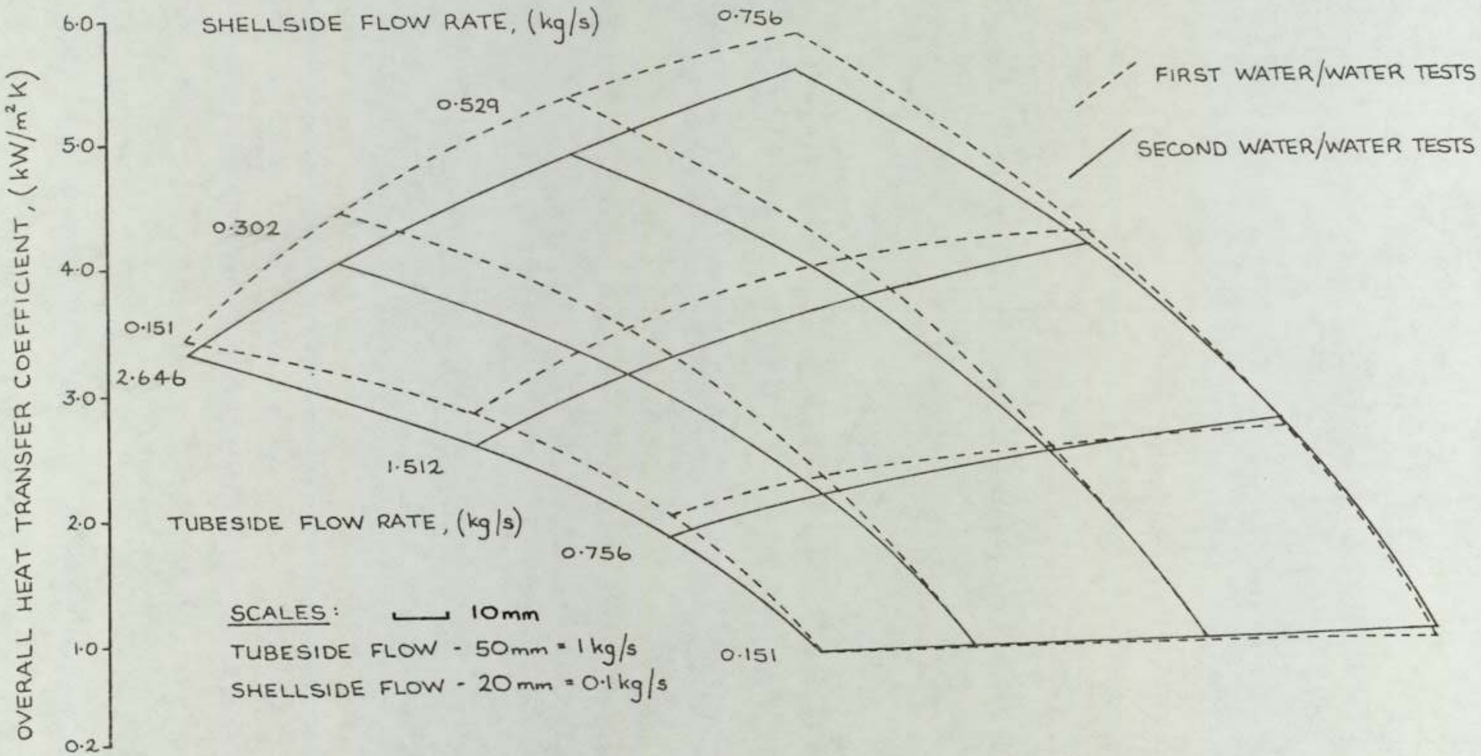


FIG. 3.6 COMPARISON OF FIRST AND SECOND WATER/WATER TESTS.

CHAPTER FOUR

EXPERIMENTAL WORK PART TWO - DYNAMIC TESTS

4.1. INTRODUCTION

The objective of these tests was to indicate whether or not the principle of dynamic testing as described in Chapter Two could be successfully applied to the testing of heat exchangers made by Serck. Experimental work was carried out for the water to water and oil to water heat exchange cases using the same heat exchanger as in the steady state tests described in Chapter Three and operated in counterflow.

To excite the dynamics of the heat exchanger a temperature and/or flow disturbance could be applied to either or both of the shell and tubeside fluids. To simplify testing requirements a single input disturbance was to be considered. Temperature forcing in preference to flow forcing was chosen because the temperature changes resulting from temperature disturbances are very much greater than those due to flow disturbances. Ideally, it is desirable to maintain the flow rate constant on the side of the heat exchanger to which the disturbance is applied, however, in practice due to the change in fluid properties during the temperature transient and its effect on the pump and fluid circuit characteristics, this is not achieved.

The temperature disturbance applied was a positive step change to the shellside fluid. Time did not permit the application of a negative step input or the application of the disturbance to the tubeside fluid in order to develop further a model of the heat exchanger. Two temperature step disturbances of magnitude 30K (20 - 50^{o}C) and 15K (35 - 50^{o}C) were chosen with the intention of examining the effect of the magnitude of the step on the mathematical model to be developed and the subsequent steady state predictions. The step input in temperature was applied by switching the flow from a supply tank containing 'cold' fluid to one containing fluid at a higher temperature. However, due to losses within the fluid and the heat transfer occurring as the disturbance travels between the supply tanks and the heat exchanger, the temperature step is degraded and, therefore, the input to the test unit is not a true step.

The change in flow rate during the temperature transient constitutes an additional input disturbance to the heat exchanger which should be recorded to enable its effect to be included in the model. Unfortunately, suitable flowmeters to measure the flow transient were not available in the laboratory, nor was there money available to purchase them. An attempt was made to give an indication of the nature of this change in flow rate and is described in a later section.

4.2. INSTRUMENTATION

4.2.1. TEMPERATURE MEASUREMENT

The fluid temperatures at the test heat exchanger's inlets and outlets were measured by thermocouples connected together to form thermopiles. Each thermopile comprised four welded bare bead measuring junctions connected in series and made from 0.2 mm diameter Copper-Constantan thermocouple wire. This arrangement gave an output of approximately 0.160 $mV/^{o}C$. The four reference junctions were supported by a PVC stopper in a wide necked Thermos Flask of 1 litre capacity, containing a mixture of melting crushed ice and water. To ensure that the reference junction was maintained at $0^{o}C$ the mixture was stirred regularly and a mercury in glass thermometer inserted with its bulb at the bottom of the Flask to detect water at a temperature slightly above $0^{o}C$. The construction of each thermopile is shown in fig. 4.1 whilst photograph 2 pictures the four thermopiles. For positioning of the thermopiles relative to the test unit see section 4.3.1.

4.2.1.1. THERMOPILE CALIBRATION

The thermopiles were calibrated in a constant temperature bath over the range 5 - $80^{o}C$ in $5^{o}C$ increments against a class 'A' mercury in glass thermometer. A 4th order calibration equation recommended in (5) and of the form:

$$T = A + BV + CV^2 + DV^3 + EV^4$$

where T = Temperature (^{o}C)

V = e.m.f. (mV)

A,B,C,D,E = constants

was used to give an individual calibration equation for each thermopile. The constants A,B,C,D,E were evaluated using the curve fitting routine POLFIT in the Honeywell STATSYST*** package.

The maximum calibration uncertainty was assumed to consist of two components which are added together. First, the scatter of the data points about the calibration curve which was $\pm$ 0.05K , and second, the uncertainty in reading the calibration thermometer of $\pm$ 0.05K. This gave an overall maximum uncertainty of $\pm$ 0.1K.

As the oil/water tests were conducted approximately 9 months after the water/water tests a series of 4 spot checks on the calibration of each thermopile was carried out. Agreement of the spot checks with the original calibration equations was better then $\pm$ 0.05K. Therefore, it was concluded that the calibrations had not altered. The calibration

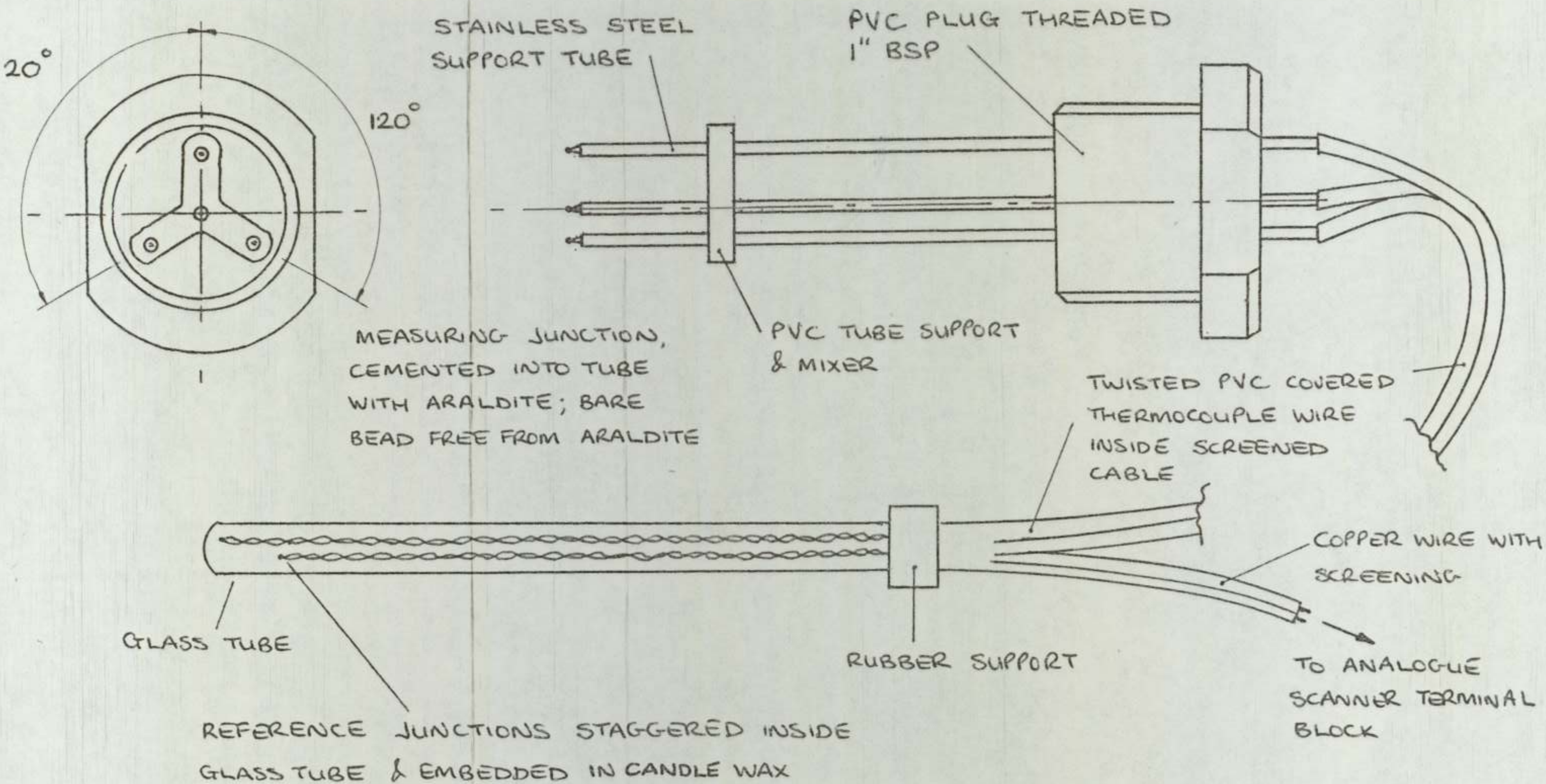


FIG. 4.1 THERMOPILE CONSTRUCTION.

equations are given in Appendix A4.

4.2.1.2. ESTIMATE OF THE DYNAMIC RESPONSE OF THE THERMOPILES

To estimate the dynamic response of the thermopiles a step input in temperature was imposed on each of the thermopiles and the response recorded on an oscilloscope. The response was assumed to be described by the response of a first order system and the time constant calculated from measurements taken from the oscilloscope trace; as shown in fig. 4.2.

For the dynamic response of the thermopiles in water, the step input in temperature was imposed by quickly transferring the thermopile by hand from a beaker of water at $0^{o}C$ to another at approximately $80^{o}C$. This procedure was repeated for several intermediate temperatures and for negative step inputs, e.g. 80 - $0^{o}C$. From these tests the time constant was estimated to be about 40 ms.

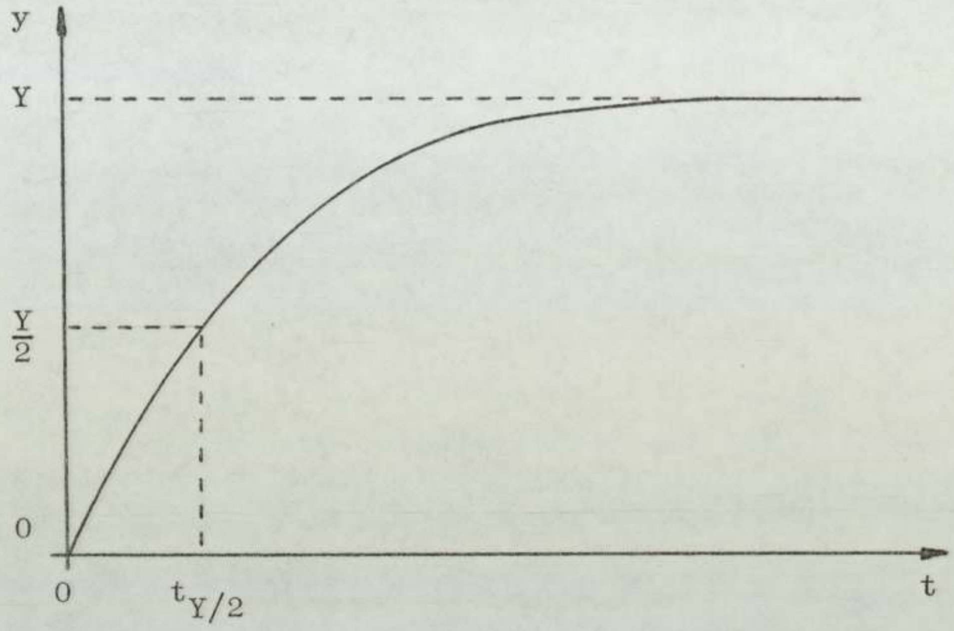
A similar series of tests were performed to estimate the time constant for the thermopiles in oil, except that the step input was made from ambient air at about $15^{o}C$ to oil at $60^{o}C$. The resulting time constant was about 120 ms.

4.2.2. FLOW MEASUREMENT

Water mass flow rates were measured using orifice plates in orifice units with D & D/2 pressure tappings, manufactured to BS 1042 Part 1: 1964. The calibration curve was generated from the British Standard and verified by dead weight measurement. Pressure differentials across the orifice plates were measured with mercury 'U' tube manometers.

Oil mass flow rates were measured with a variable area type flowmeter calibrated by dead weight measurement over the

FIG. 4.2. DETERMINATION OF THERMOPILE TIME CONSTANT



Assuming thermopile response is described by a first order system:

$$y = Y(1 - e^{-t/T})$$

where T = system time constant

Now, if $y = \frac{Y}{2}$, then

$$\frac{y}{Y} = \frac{Y}{2Y} = 1 - e^{\frac{-t_{Y/2}}{T}}$$

$$\text{hence } T = \frac{t_{Y/2}}{0.6931}$$

range 0.113 - 0.945 kg/s at 20 and 50^{o}C.

4.2.3. DATA COLLECTION

The thermopile voltage readings were recorded via a Solartron Data Logging System and output onto papertape by either a Facit Tape Punch or a Data Dynamics Teletype Tape Punch. The logging equipment was suitably shielded from any draughts of cool air and the thermopile input terminals encased in polystyrene to maintain them at an equal temperature. Details of the Data Logger are given in Appendix A4.

The remaining test variables were recorded by hand.

4.3. TEST RIG DEVELOPMENT

4.3.1. ASPECTS COMMON TO BOTH THE WATER/WATER AND OIL/WATER TEST RIGS

The principle of dynamic testing has previously been described. In order to reduce the degradation of the temperature step input disturbance applied to the shellside of the test heat exchanger, the amount of metal between the supply tanks and the shell inlet to absorb heat was kept to a minimum by using ABS pipe fittings where possible.

It was necessary to operate the fluids on open circuit because of the temperature changes occurring during the tests.

The change in shellside flow rate caused by the decreasing supply head was minimised by using a positive displacement pump.

The initial shell flow rate and the 'hot' supply tank temperature were arranged such that the steady state conditions prevailing after the application of the temperature input, corresponded approximately to the test conditions in

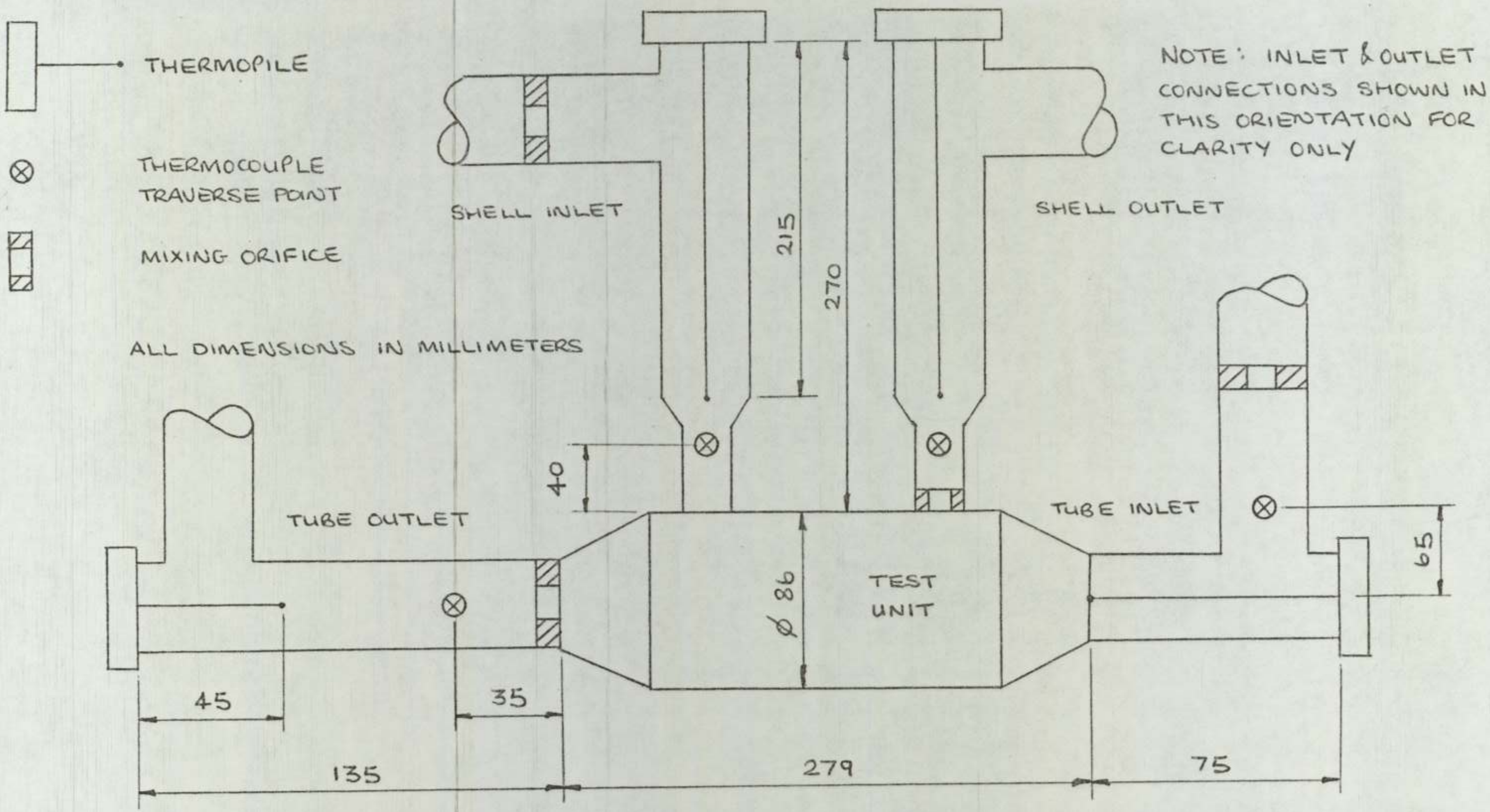


FIG. 4.3 POSITION OF THERMOPILES AT TEST UNIT INLET AND OUTLET PORTS.

the conventional steady state tests.

The thermopiles were screwed into the end of the 'Tee' piece ABS fittings and positioned relative to the test unit as shown in fig. 4.3. Mixing orifices were placed upstream of the measuring junctions to promote good fluid mixing. Pipe side tappings were positioned close to the thermopile measuring junctions to enable a thermocouple to be traversed across the pipe diameter to establish if any temperature gradients existed, which would indicate incomplete fluid mixing. A thermocouple traverse at each end of the thermopile measuring points was carried out for the minimum and maximum flow rates and temperatures to be encountered during the tests for each fluid. The temperature traverses indicated that negligible temperature gradients existed across the pipes and that the fluid temperature being measured by the thermopiles was a bulk or mixing cup temperature.

The test unit and the pipework in the vicinity of the thermopiles were well lagged to minimise heat losses.

4.3.2. WATER/WATER TEST RIG

The water/water test rig was the first rig to be constructed. A circuit diagram is shown in fig. 4.4 and the equipment used detailed in Appendix A4.

The 'hot' and 'cold' water supply tanks were insulated and the surface of the water in the 'hot' tank covered with Allplas plastic spheres to minimise evaporation losses. The water in the two tanks was heated via a common steam heater and mixed in the tanks by a hand held stirrer. The temperature of the supply tank water was measured with mercury in glass thermometers suspended in the water. Both the shell and tubeside water was discharged to waste into the laboratory

FOR CODE LETTER IDENTIFICATION
SEE APPENDIX A4.2.1
FOR KEY TO SYMBOLS SEE FIG. 3.2

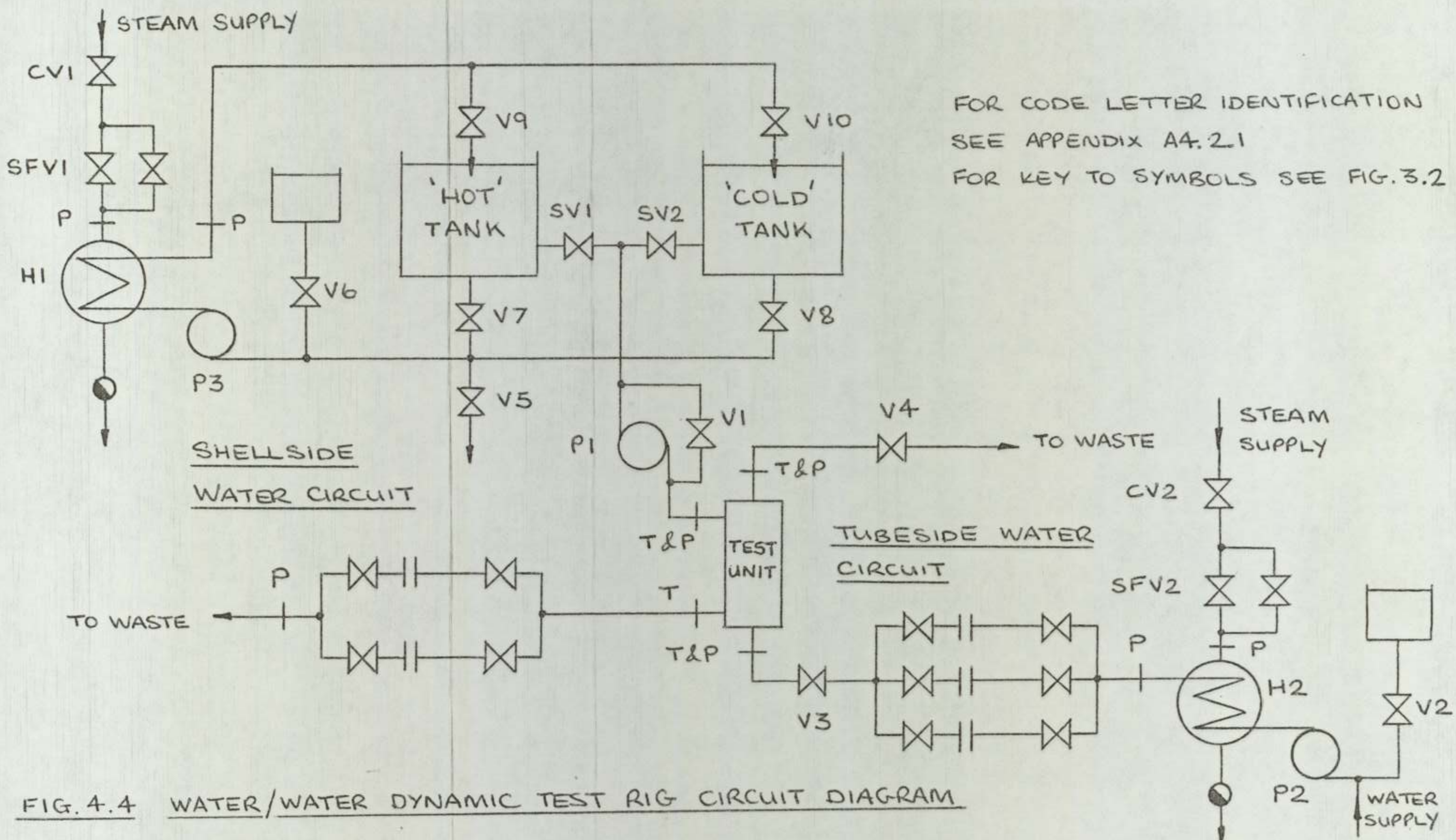


FIG. 4.4 WATER/WATER DYNAMIC TEST RIG CIRCUIT DIAGRAM

water trenches. Make up water was provided from the laboratory header tank. The shellside orifice units were placed downstream of the test unit to reduce the amount of metal in the circuit before the shell inlet and the length of piping from the supply tanks to the test unit. This enabled a temperature input as severe as practically possible to be applied to the test unit. The tubeside water was heated by a steam heater.

4.3.3. OIL/WATER TEST RIG

The oil used in these tests was Shell Diala B; a transformer oil. This was chosen because at the temperatures specified for the dynamic tests the viscosity of the oil frequently used for heat transfer tests in the laboratory would be too high, such that the oil pump available would not be able to provide the required flow rates. It was therefore necessary to choose an oil which would enable the required flows to be obtained with the existing pump and upon completion of the tests would be of use to the laboratory. Transformer oil met these requirements.

A schematic diagram of the test rig is given in fig. 4.5. Four oil tanks were provided - two supply tanks and two collection or dump tanks. The supply tank solenoid valves were linked to the dump tank solenoid valves so that the hot supply and hot dump tank valves were open while the cold supply and cold dump tank valves were closed and vice versa. An additional switching arrangement allowed the solenoid valves to be operated individually to enable oil to be redistributed between different tanks in the oil circuit.

Oil from the hot supply tank was initially heated by the steam heater and mixed in the hot dump tank. On reaching the desired temperature, the oil was pumped from the hot dump to the hot supply tank and the valves adjusted to allow the hot supply pump to circulate oil through the supply tank only. This ensured good mixing. The final desired hot tank oil temperature was achieved using electric immersion heaters controlled by a Variac. However, on start up or when the oil trapped in the bottom of the supply tank has been stationary for sometime, this layer of oil does not become mixed with the oil pumped up from the dump tank. In these situations it was necessary to half fill the supply tank, then allow the oil to drain under gravity back into the dump tank to flush out the cooler oil.

'Cold' oil to establish the initial steady state from which the temperature disturbance was applied was heated and mixed by circulating oil over electric immersion heaters in the 'cold' supply tank. An oil cooler was included in the circuit to cool oil supplied from the 'cold' dump tank.

The four oil tanks were provided with lids and were fully insulated. The temperature of the oil in the tanks was measured with Chromel/Alumel thermocouples and displayed digitally at the control panel.

The oil flowmeter was positioned downstream of the test heat exchanger to ensure that the test unit was kept in the high pressure region to assist with bleeding air from this part of the oil circuit.

On commissioning the rig it was found that the main oil pump was producing a temperature rise of up to 4K across the pump. Since it was required to attain a steady state temperature of 50°C at the test unit shell inlet after the temperature transient, it was necessary to determine experimentally the temperature rise across the pump at the required test oil

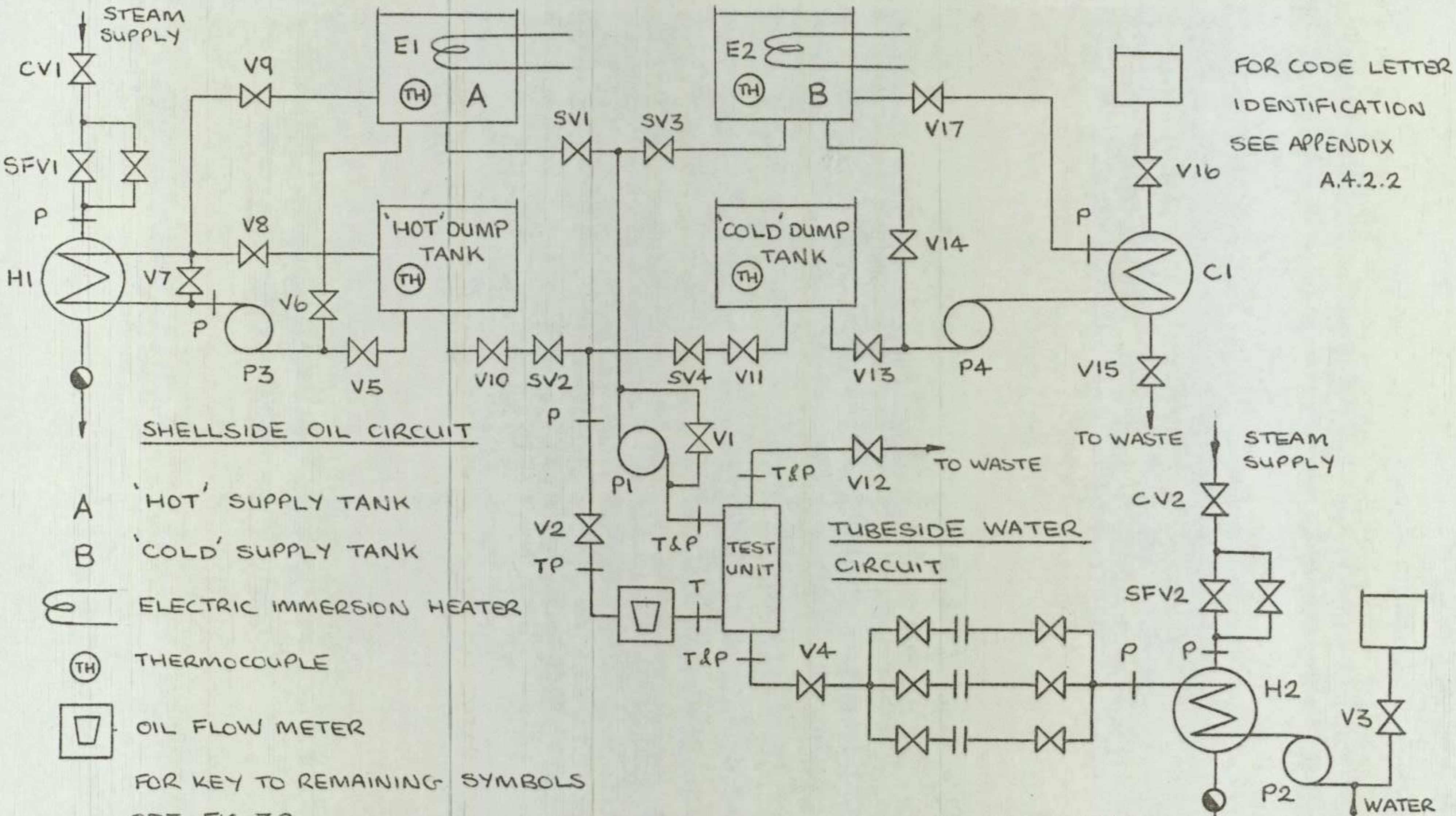


FIG. 4.5 OIL/WATER DYNAMIC TEST RIG CIRCUIT DIAGRAM.

flows. To compensate for this additional heat input the 'hot' and 'cold' supply tank oil was heated to a correspondingly lower temperature.

The tubeside water circuit was the same as for the water/water test rig.

Photographs of the oil/water test rig are given in Appendix A4.

4.4. TEST PROCEDURES

4.4.1. DURATION OF TESTS AND SAMPLING RATE

Before the tests to collect transient response data could commence, it was first necessary to estimate the duration of each test run and determine a suitable sampling rate.

The length of each test run was a compromise between the need to ensure that the test unit inlet and outlet temperatures reached a steady state condition following the applied temperature disturbance; the capacity of the supply tanks and hence the volume of fluid required and the problem of maintaining the 'hot' and 'cold' tank fluid at a constant temperature.

Tests at each combination of flow rate extremes for each magnitude of temperature step for the water/water case showed that a steady state condition had been reached 3 minutes after the application of the temperature disturbance. The steady state was indicated by successive thermopile voltage readings showing negligible variation. Similar tests for the oil/water case showed that 6 and 9 minutes was required for a steady state condition to be reached for the maximum and minimum shell flow rates respectively. These timings were used as the basis for terminating each test run.

For the water/water tests the Data Logger Scanner controls were set to sample continuously. The sampling rate was determined by initiating a continuous scan of the four thermopiles over a period of 10 minutes and then counting the number of samples recorded on the paper tape. This gave a sampling rate of 0.3s between successive samples and 1.2s between samples of the same variable.

During the oil/water tests two different recording devices were used. For the Data Dynamics Teletype Tape Punch similar tests to those described above, except that the scan was timed over 20 minutes, gave sampling rates of 1.4s between successive samples and 5.6s between samples of the same variable. When the Facit Tape Punch was used, the sampling rate between successive samples was 1.2s while the scan rate for successive samples of the same variable was set at 2.0s. These sampling rates were verified in a similar manner to those previously described.

4.4.2. TEST PROCEDURE FOR DYNAMIC TESTS

4.4.2.1. TEMPERATURE DISTURBANCE APPLIED TO SHELLSIDE FLUID

1. The tubeside water circuit was first bled of air, then the required flow rate set. The tubeside inlet temperature was adjusted to $20^{\circ}C \pm 1K$ and allowed to stabilize. The temperature was indicated by the Data Logger DVM.
2. The fluid in the 'hot' and 'cold' supply tanks was heated to the appropriate temperature and any entrained air present allowed to settle out.
3. When steps 1 and 2 were satisfied fluid was drawn from the 'cold' supply tank, air bled from the circuit and the initial shellside flow rate set. The test unit inlet and outlet temperatures were allowed to stabilize - the thermopile voltage readings displayed on the DVM were used to indicate this.

4. When steady state conditions were reached data collection was initiated (Data Logger switched to scan) and the initial steady state data collected for about 30s. During this time the initial shellside flow rate was recorded.

5. At the end of this 30s period the positive step temperature disturbance was applied by closing the 'cold' supply tank outlet solenoid valve and opening the 'hot' supply tank outlet solenoid valve.

6. OIL/WATER TESTS ONLY:

   To give an indication of the change in oil flow rate with time following the applied temperature disturbance, a stop watch was started at the same moment as the step input was applied and the flow indicated by the Rotameter recorded at suitable time increments.

7. The tubeside inlet temperature was required to be constant. If it was observed to fluctuate significantly due to steam disturbances during a test run, the test was repeated.

8. Prior to terminating the test run, the tubeside water flow rate and the final steady state shellside flow rate were recorded.

9. Each test was terminated as follows:-

   WATER/WATER TESTS: At the end of 3 minutes following the step input, a negative step temperature input was applied by switching the fluid supply from the 'hot' to the 'cold' supply tank. Data was collected for a further 2 minutes and the test then terminated.

   OIL/WATER TESTS: After the specified time to reach steady state conditions had elapsed, data collection was halted and a negative step temperature input applied. The test was then terminated. (The purpose of imposing the negative steps was to force the fluid and circuit metal temperatures down towards the initial steady state temperature levels from which the temperature disturbance is applied. This reduced the time required to reach the initial steady state for the next test run).

10. The above procedure was repeated for the next test run.

4.4.2.2. STEP INPUTS IN FLOW APPLIED TO THE SHELLSIDE FLUID, (OIL/WATER TEST ONLY)

Details of the step inputs in flow applied are given in Table 4.1.

1. The water flow rate was set as described in section 4.4.2.1, step 1
2. The oil in the 'hot' supply tank was heated to the required temperature.
3. With solenoid valves SV1 and SV2 open, with SV3 and SV4 closed, the nominal oil flow was set by regulating the pump by-pass valve V1, (fig. 4.5).
4. SV4 was then opened and V11 adjusted to give the required magnitude of step input in flow.
5. Conditions were then allowed to stabilize.
6. Data collection was initiated as per section 4.4.2.1, step 4.
7. 30s into the data collection, the step input in flow was applied by closing SV4. (The transient effects had disappeared and steady state conditions reached after 3 - 4 minutes).
8. The change in flow with time was recorded by visual observation and stop watch.

9. Upon reaching the final steady state the test was terminated and the procedure repeated for the next test point.

4.5. TEST SUMMARY

The dynamic tests consisted of applying a step in temperature to the shellside fluid from a initial steady state condition while maintaing the tubeside inlet temperature constant. The tests were conducted at the same nominal shell and tubeside flow rates as those in the conventional steady state tests.

Temperature steps of magnitude 30K and 15K corresponding to the temperature changes of 20 - 50^{o}C and 35 - 50^{o}C were chosen. These tests were carried out for the fluid combinations of water to water and oil to water heat exchange.

A series of tests in which a step change in flow was applied to the shellside fluid in the oil/water tests was also performed.

A summary of the test content for the dynamic tests is given in Table 4.1.

TABLE 4.1. SUMMARY OF DYNAMIC TESTS

WATER/WATER TESTS

Shell and tubeside fluids : water

OIL/WATER TESTS

Shellside fluid : oil (Shell Diala B)

Tubeside fluid : water

TEST CONDITIONS:

Final steady state shellside fluid inlet temperature (nominal): 50^{o}C

Nominal tubeside fluid inlet temperature: 20^{o}C

Temperature step disturbances (nominal) of 30K (20-50^{o}C) and 15K (35-50^{o}C) applied to shellside fluid

(a) STEP INPUT IN TEMPERATURE TESTS

		Nominal Tubeside Water Flow Rate (kg/s)			
		0.151	0.340	0.756	1.512
NOMINAL SHELLSIDE FLUID FLOW RATE (kg/s)	0.151	X 0 □	X □	X 0 △	X △
	0.302	X 0	X △	X	X □
	0.529	X 0	X 0	X 0	X 0
	0.756	X 0 △	X 0	X 0 □	X

X - Test condition for all tests

Repeat Points: Water/Water, 30K step - 0

Water/Water, 15K step - None

Oil/Water, 30K step - △

Oil/Water, 15K step - □

(b) STEP INPUT IN FLOW TESTS (OIL/WATER TEST ONLY)

Negative step inputs in flow applied to the shellside fluid, i.e. the input causes a decrease in flow.

The change in flow is expressed as:-

$$\frac{(\text{Initial Flow} - \text{Nominal Flow})}{\text{Nominal Flow}} \times 100 \quad \%$$

(i) OIL INLET TEMPERATURE $50^{\circ}C$

Nominal Oil Flow Rate (kg/s)	Nominal Water Flow Rate (kg/s)	
	0.151	1.512
0.151	-15 & -4%	-15 & -4%
0.756	-5 & -2%	-5 & -2%

(ii) OIL INLET TEMPERATURE $35^{\circ}C$

Nominal Oil Flow Rate (kg/s)	Nominal Water Flow Rate (kg/s)	
	0.151	1.512
0.151	-15%	-15%
0.756	-5%	-5%

CHAPTER FIVE

ANALYSIS AND DISCUSSION OF DYNAMIC TEST RESULTS

It has already been explained that there was insufficient time available to analyse the large amount of experimental data collected, in order to assess the feasibility of predicting the steady state thermal performance of a heat exchanger from measurement of part of its transient response. For this reason the analysis which follows is limited to the first few steps in the development of a model to describe the dynamic characteristics of the water to water heat exchanger. It is based on the consideration of six test points for the 30 K step input disturbance - four at the extremes in flow rate and two in the mid flow range.

5.1. ANALYSIS OF RESULTS

Table 5.1 gives a summary of the test details for the test runs analysed. (Note: the flow rates given previously in Table 4.1 were only nominal, not actual flow rates).

The analysis makes use of the suite of programmes described in Chapter Two.

A typical output record showing the interactive features of the first programme, SORT, is given as Appendix A5.1. The temperature responses for runs 1 & 7 are shown through figs. 5.1 - 5.6.

The output from programme DFTR for runs 1 & 7 are given as Appendix A5.2. This output is typical for the data processed. Note that a frequency response analysis was not performed on the tubeside inlet temperature because the temperature was maintained essentially constant during the tests. A problem which occurred with the DFTR routine is clearly illustrated in the error signal frequency response data for the shell inlet temperature in run 7 and the shell outlet temperature in both runs 1 & 7 (Appendix A5.2). The response seems well approximated by a first order response (see later), however, below 0.02 rad/s the phase angle and amplitude are not consistent with this premise. This discrepancy in the frequency response data was apparent in the other test runs processed.

Due to possible errors in the error signal frequency response data at low frequency, the minimum frequency for which data was generated by programme DFTR for subsequent processing by programme FITFRQ was 0.05 rad/s. From the Bode plots the amplitudes and phase angles at frequencies <0.05 rad/s were estimated and inserted in the frequency response files. Figs. 5.7 - 5.12 show the Bode diagrams for runs 1 & 7. Curve fitting by programme FITFRQ was over the range $0.002 \leq \omega \leq 100$ rad/s. The number of programme iterations required to give satisfactory convergence of the polynomial coefficients was determined for several combinations of poles and zeros by comparing the coefficients after different numbers of iterations. The result was that a minimum of 5 iterations should be used.

Although the Bode diagrams indicated a first order response (i.e. single pole) combinations of different numbers of poles and zeros were tried. For a fit using 1 zero and 2 poles the results for different test runs were inaccurate and inconsistent. When converted into the time domain the early part of the transient was often in significant error. In some cases a positive pole resulted indicating a growth exponential in the time domain solution which in reality could not be correct. The analysis has, therefore, been restricted to considering the error signal response as first order. A specimen output from programme FITFRQ is given as Appendix A5.3.

Table 5.2 shows a comparison for runs 1 & 7 between the 'true' error signal amplitude and the amplitude estimated by hand calculation using the -3db & 45^{0} phase points in the Bode plots and the amplitude determined in programme FITFRQ. Table 5.3 compares the 'true' error signal amplitude with that calculated in FITFRQ for the remaining test runs - 6, 14, 15 and 16.

Should the incorrect error signal amplitude be carried forward into the evaluation of C(s) a zero will appear in the expression for C(s) - see specimen output from programme ESTOCS, Appendix 5.4. When the correct amplitude is used a zero will not be present.

Table 5.4 presents expressions for C(s) incorporating the 'true' error signal amplitudes, transport lags and initial conditions. For runs 1 & 7 the time constants have been taken from Table 5.2 for the -3db point, while for the other runs those determined in programmes FITFRQ/ESTOCS have been used. The zero resulting from the incorrect error signal amplitude being used in ESTOCS has been omitted.

Superimposed on figs. 5.1 - 5.6 is the first order response for which the time constant has been determined from the appropriate Bode plot assuming the 'true' error signal amplitude and the Bode diagram amplitude in decibels.

5.2. DISCUSSION OF RESULTS

The temperature transient plots for runs 1 & 7 clearly show a typical 'S' shape response curve characteristic of a multi-exponential process. The superimposed first order response curves indicate that the true temperature responses are not well approximated by a single time constant response.

The frequency versus amplitude/phase plots for the corresponding error signals show a distinct 90^{o} phase lag at frequencies $>$1-5 rad/s indicating that over this frequency range the process is behaving as a first order system. The slight overshoot exhibited at low frequencies by the error signal amplitude crossing the asymptote is further evidence that the responses are not simple first order responses as assumed in the analysis.

An attempt was made to obtain a better approximation for the transient response by considering the error signal frequency response to be described by 1 zero and 2 poles, however, when inverted into the time domain serious errors resulted. The inaccuracies stem from the raw experimental data and the generation of the frequency response, not from deficiencies in the curve fitting algorithm, since the latter has been used extensively by other researchers and has given satisfactory results. The errors in the error signal frequency response data at low frequencies have already been discussed in the analysis section. The result was to restrict the minimum frequency for evaluation by programme DFTR to 0.05 rad/s and estimate by hand the lower frequency data. Consequently, reliable low frequency data was not available. Accurate data in this region would be essential in order to curve fit to the overshoot mentioned above and to give accurate time domain data when the resulting expressions are inverted. Reference has also been made in section 2.3.2 to the restriction on the maximum data sampling rate which could be used. The estimation of the response parameters can be particularly sensitive to the data over the early part of the transient since many points may be needed to adequately describe the response. Consequently, increasing the number of samples, for example by interpolating between the existing experimental points, would be expected to give improved results. From Table 5.1 for runs 6 & 7, the 'tube out' transport delays for the same tube flow rate are significantly different, therefore, a re-appraisal of the approximations made in section 2.3 to estimate the start of the transient response which will define the content of the early part of the response may be necessary.

Table 5.1 also shows that the tube inlet temperature was maintained reasonable constant, although it is well known that a fluctuation of only 0.1K will produce a fluctuation in the outlet temperatures. Thus, the responses could contain a slight ripple or process noise.

Tables 5.2 and 5.3 highlight another of the analysis problems which again probably result from inadequate data. The tables compare the 'true' error signal amplitude and values obtained by several methods. The agreement between the 'true' error signal amplitude and that reconstructed using equation 2.12 varies from good to poor. Generally, the amplitude estimated by hand utilising the -3db point gives closer agreement.

From the limited analysis summarised in Table 5.4 a few trends can be established. As would be expected for increasing shell flow rate the shell inlet temperature time constant decreases and is of course independent of tube flow rate. Over the shell flow range 0.147 - 0.756 kg/s the shell inlet temperature time constant varied between 10.42 and 2.10 s respectively. The shell outlet temperature time constant varied similarly, but as expected was slightly longer than the inlet time constant, 11.63 - 2.28 s. However, it is to be expected that the shell outlet time constant is also dependent on the tubeside flow rate. The tube outlet temperature time constant decreased for both increases in shell & tube flow rates, (12.19 - 2.52s). Over the tube flow range 0.150 - 0.756 kg/s the two outlet temperatures only showed a slight depdence on the tube flow rate compared to the strong dependence shown to shell flow rate.

It is interesting to note that for some cases when the transfer function relationships are taken the numerator and denominator time constants almost cancel each other, yielding an expression containing only the amplitude ratio attenuation and the transport delay terms. This implies that after the transport delay has ended the temperature difference between the inlet and outlet is almost constant.

Had circumstances permitted it was intended to develop a model describing the heat exchanger's dynamic behaviour of the form illustrated in fig. 5.13 and incorporate additional elements at a later stage to relate changes in shell flow rate to the outlet temperatures. For the water/water case the changes in shell flow rate during the transient were only slight (Table 5.1), however, for oil to water heat exchange the changes were considerable, hence the experimental work in section 4.4.2.2.

A further aspect of the work would have been to relate the time constants and gains in the model to the physical parameters of the heat exchanger - flow rates, fluid properties, metal mass etc. It is well known that simple transfer functions of the form:

$$G(s) = \frac{K \prod_{m} (1 + T_m s)\, e^{-T_d s}}{\prod_{n} (1 + T_n s)} \qquad (5.1)$$

comprising only a few numerator and/or denominator elements can adequately describe the frequency response behaviour of a heat exchanger. Cahn and Leonard (8) followed by Aly (1) have started the work of relating the physical parameters of the heat exchanger to the model parameters given in equation 5.1. Further research following on from their work is needed.

TABLE 5.1 SUMMARY OF TEST DETAILS FOR THE WATER/WATER TEST POINTS ANALYSED.

RUN NO.	MAGNITUDE OF TEMP INPUT (K)	TUBESIDE FLOW RATE (kg/s)	*SHELLSIDE FLOW RATE (kg/s)			TRANSPORT DELAY (s)†			TUBE INLET TEMP VARIATION (°C) □		NO. OF DATA POINTS ANALYSED ●		
			BEFORE	DURING	AFTER	SHELL IN	SHELL OUT	TUBE OUT	MIN	MAX	TH1	TH2	TC2
1	26.84	0.151	0.147	0.147	0.147	9.04	3.9	5.7	19.78	19.87	144	141	140
6	29.50	0.756	0.756	0.785	0.756	1.70	0.3	0.9	19.78	20.02	129	129	129
7	28.90	0.756	0.150	0.149	0.150	8.96	2.7	3.3	20.13	20.41	110	108	108
14	29.99	0.341	0.517	0.529	0.517	2.52	0.3	2.1	19.90	19.97	136	136	135
15	29.30	0.343	0.287	0.302	0.286	4.41	1.5	2.1	19.99	20.04	144	143	143
16	29.58	0.150	0.724	0.750	0.718	1.78	0.3	2.1	20.50	20.68	147	147	146

* BEFORE ~ flow rate prior to temperature disturbance being applied
DURING ~ flow rate immediately before negative temperature step applied
AFTER ~ flow rate immediately before termination of test

† estimated in program SORT
SHELL IN ~ time for temperature disturbance to travel from supply tanks to heat exchanger shell inlet
SHELL OUT ~ time for temperature disturbance to travel from shell inlet to shell outlet
TUBE OUT ~ time delay between temperature disturbance reaching shell inlet & response first indicated at tube outlet

□ determined in program SORT

● number of data points output from program SORT

TABLE 5.2 COMPARISON OF ERROR SIGNAL TIME CONSTANTS AND AMPLITUDES FOR WATER/WATER TEST RUNS 1 & 7 ASSUMING FIRST ORDER RESPONSE.

			TIME CONSTANT BASED ON:					
		* HAND CALCULATION FROM FIGS. 5.7 - 5.12					PROGRAM FITFRQ & EQUATION 2.12	
			- 3 db POINT		- 45° PHASE			
RUN & TEMPERATURE I.D.	TRUE ERROR SIGNAL AMPLITUDE (K)	ERROR SIGNAL GAIN (dbs)	TIME CONSTANT (S)	AMPLITUDE (K)	TIME CONSTANT (S)	AMPLITUDE (K)	TIME CONSTANT (S)	AMPLITUDE (K)
RW1TH1	26.84	49.0	10.42	27.04	10.87	25.93	11.23	31.62
RW1TH2	21.47	48.0	11.63	21.60	11.76	21.36	12.63	24.60
RW1TC2	5.69	37.0	12.19	5.81	13.33	5.31	13.92	6.44
RW7TH1	28.90	49.0	9.71	29.02	9.52	29.59	9.30	32.78
RW7TH2	16.31	45.0	10.75	16.54	11.11	16.01	10.68	18.47
RW7TC2	2.46	27.5	9.52	2.49	12.50	1.90	11.34	2.48

* at the -3db point & where the phase angle is -45° the time constant T is given by the reciprocal of the frequency at the corresponding point. The error signal amplitude is calculated from the equation, gain (dbs) = $20 \log_{10} ET$, where E = error signal amplitude.

TABLE 5.3 COMPARISON OF ERROR SIGNAL AMPLITUDES - TRUE AGAINST CALCULATED*

RUN No.	ERROR SIGNAL AMPLITUDE (K)					
	TH1		TH2		TC2	
	TRUE	CALCULATED	TRUE	CALCULATED	TRUE	CALCULATED
6	29.50	26.15	25.45	28.64	4.11	4.15
14	29.99	29.96	26.36	30.44	5.21	5.32
15	29.30	29.26	23.75	24.01	4.70	4.81
16	29.58	25.32	28.63	28.29	8.56	10.13

* in program FITFRQ

TH1 ~ shellside inlet temperature, TH2 ~ shellside outlet temperature
TC1 ~ tubeside inlet temperature, TC2 ~ tubeside outlet temperature

TABLE 5.4 C(S) EXPRESSIONS INCLUDING TRANSPORT LAGS AND INITIAL CONDITIONS FOR THE WATER/WATER TEST POINTS ANALYSED

RUN NO.	SHELL INLET TEMPERATURE (°C)	SHELL OUTLET TEMPERATURE (°C)	TUBE OUTLET TEMPERATURE (°C)
1	$\frac{26.84}{s(1+10.42s)} + \frac{20.76}{s}$	$\frac{21.47e^{-3.9s}}{s(1+11.63s)} + \frac{20.53}{s}$	$\frac{5.69e^{-5.7s}}{s(1+12.19s)} + \frac{20.93}{s}$
6	$\frac{29.5}{s(1+2.10s)} + \frac{20.65}{s}$	$\frac{25.44e^{-0.3s}}{s(1+2.28s)} + \frac{20.54}{s}$	$\frac{4.11e^{-0.9s}}{s(1+2.52s)} + \frac{20.01}{s}$
7	$\frac{28.90}{s(1+9.71s)} + \frac{21.04}{s}$	$\frac{16.31e^{-2.7s}}{s(1+10.75s)} + \frac{20.73}{s}$	$\frac{2.46e^{-3.3s}}{s(1+9.52s)} + \frac{20.42}{s}$
14	$\frac{29.99}{s(1+3.38s)} + \frac{20.65}{s}$	$\frac{26.36e^{-0.3s}}{s(1+3.71s)} + \frac{20.54}{s}$	$\frac{5.21e^{-2.1s}}{s(1+4.85s)} + \frac{20.09}{s}$
15	$\frac{29.30}{s(1+5.13s)} + \frac{21.16}{s}$	$\frac{23.75e^{-1.5s}}{s(1+5.76s)} + \frac{20.99}{s}$	$\frac{4.70e^{-2.1s}}{s(1+5.94s)} + \frac{20.21}{s}$
16	$\frac{29.58}{s(1+2.20s)} + \frac{20.65}{s}$	$\frac{28.63e^{-0.3s}}{s(1+2.89s)} + \frac{20.01}{s}$	$\frac{8.56e^{-2.1s}}{s(1+3.70s)} + \frac{20.55}{s}$

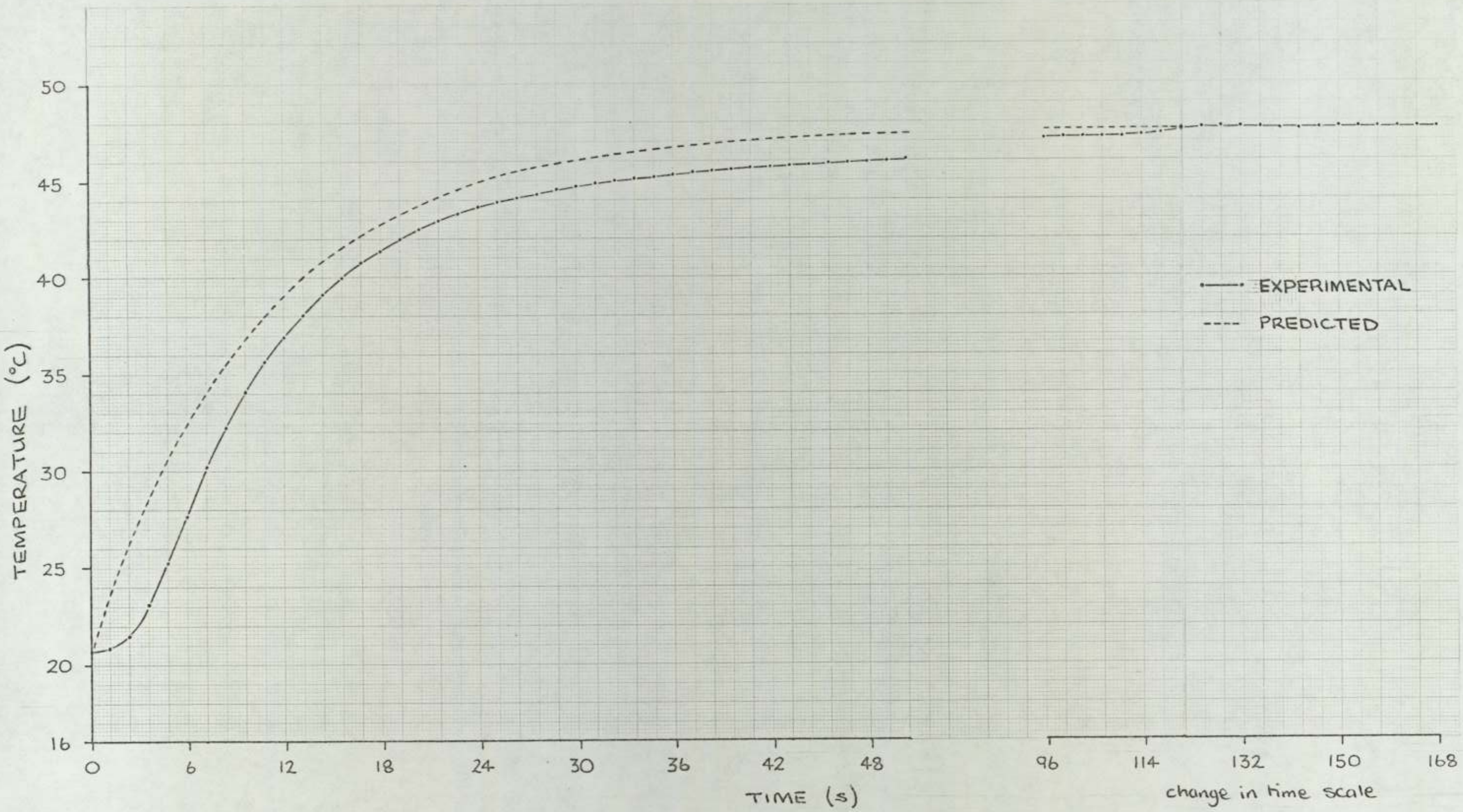


FIG. 5.1 TEMPERATURE TRANSIENT RESPONSE, RUN 1, SHELL INLET

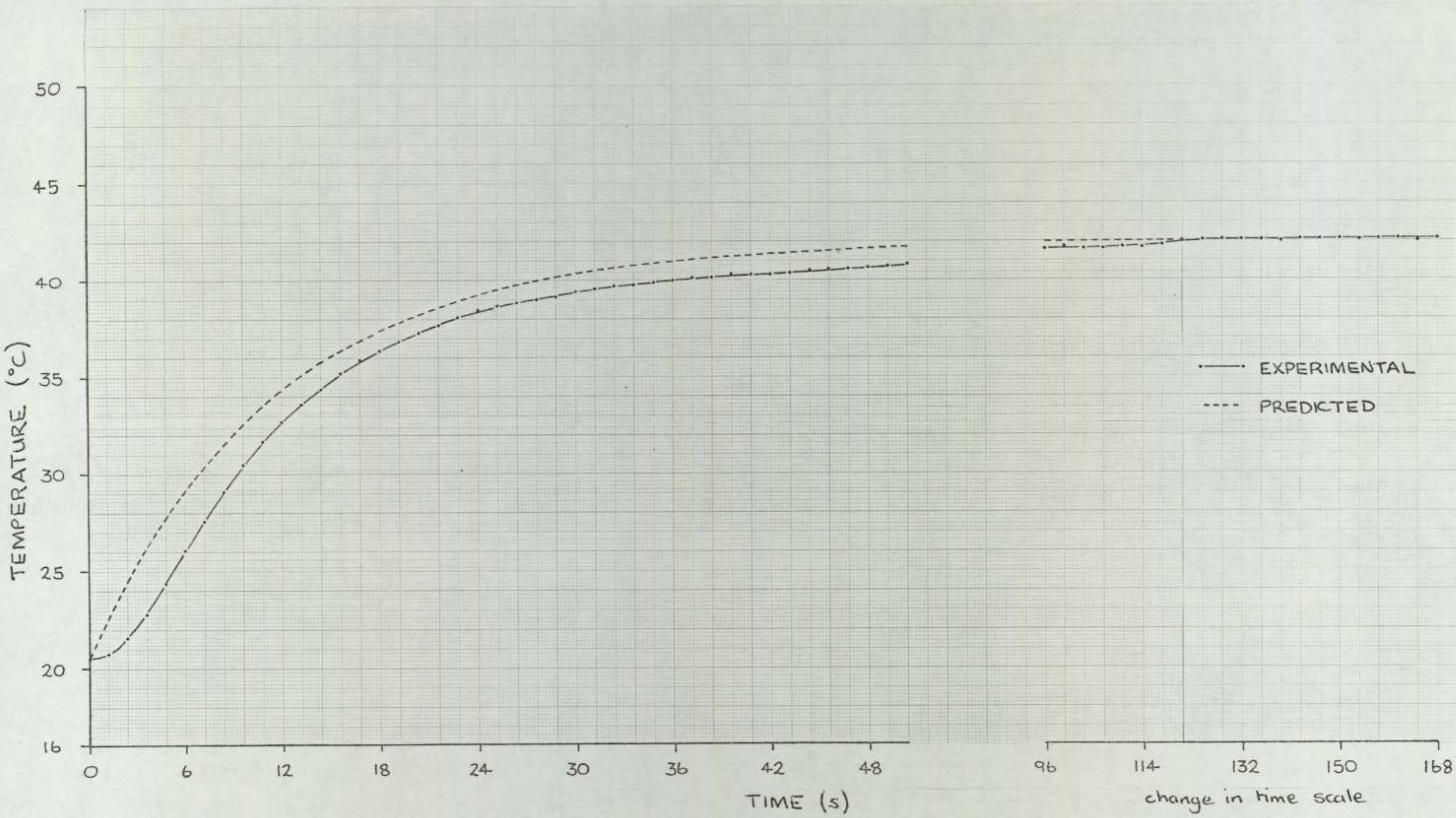


FIG. 5.2 TEMPERATURE TRANSIENT RESPONSE, RUN 1, SHELL OUTLET.

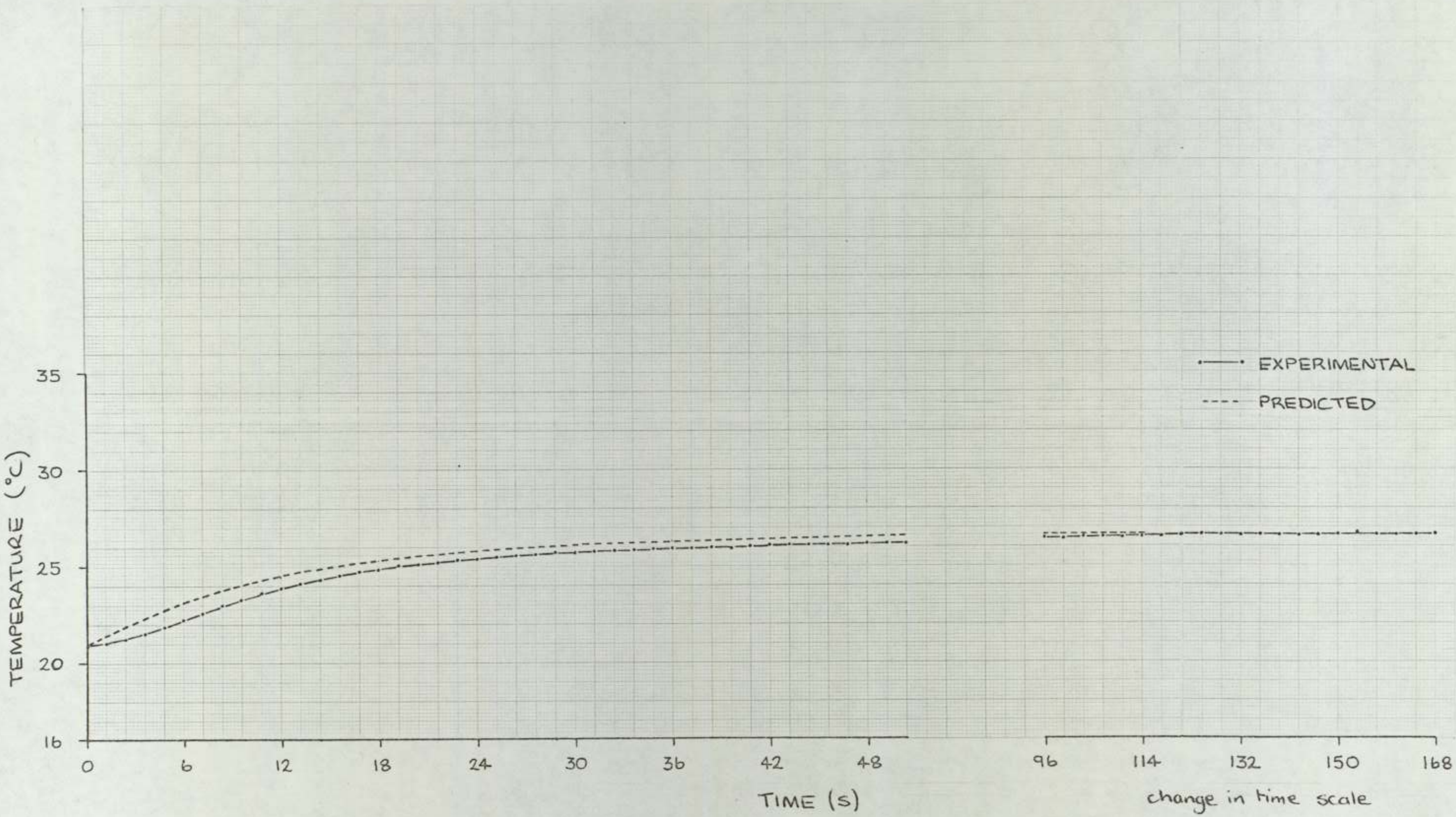


FIG. 5.3 TEMPERATURE TRANSIENT RESPONSE, RUN 1, TUBE OUTLET

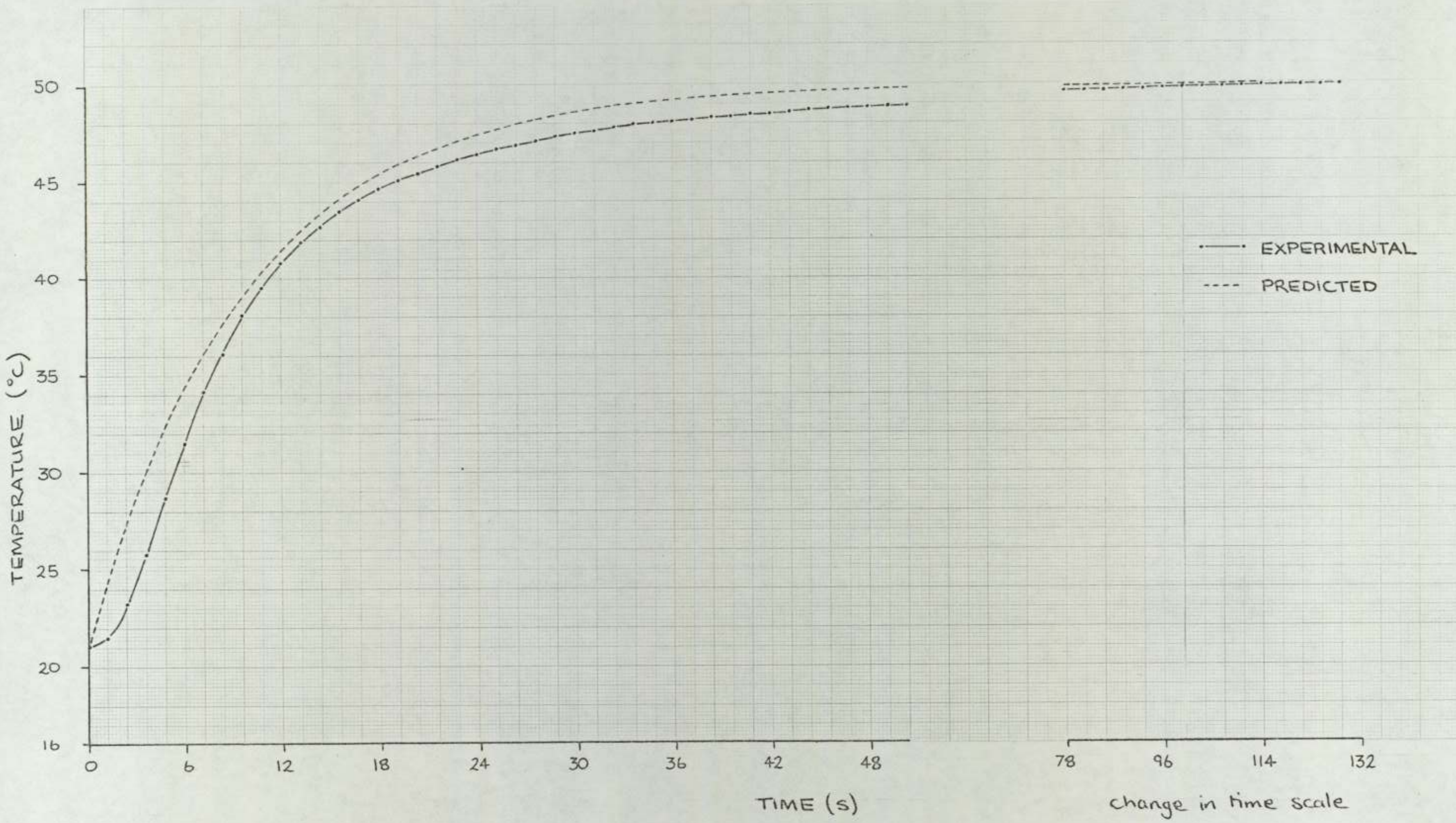


FIG. 5.4 TEMPERATURE TRANSIENT RESPONSE, RUN 7, SHELL INLET

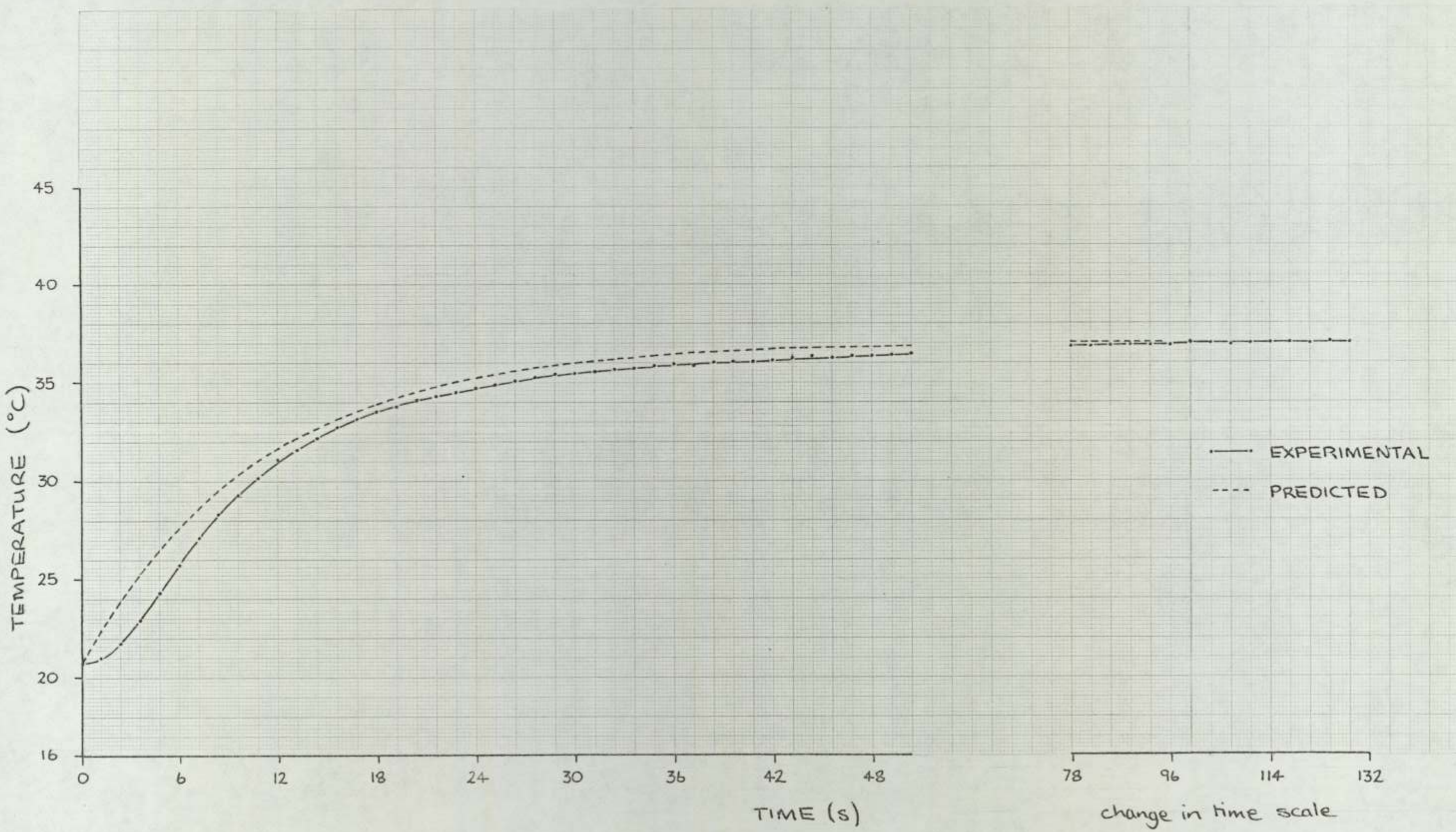


FIG. 5.5 TEMPERATURE TRANSIENT RESPONSE, RUN 7, SHELL OUTLET

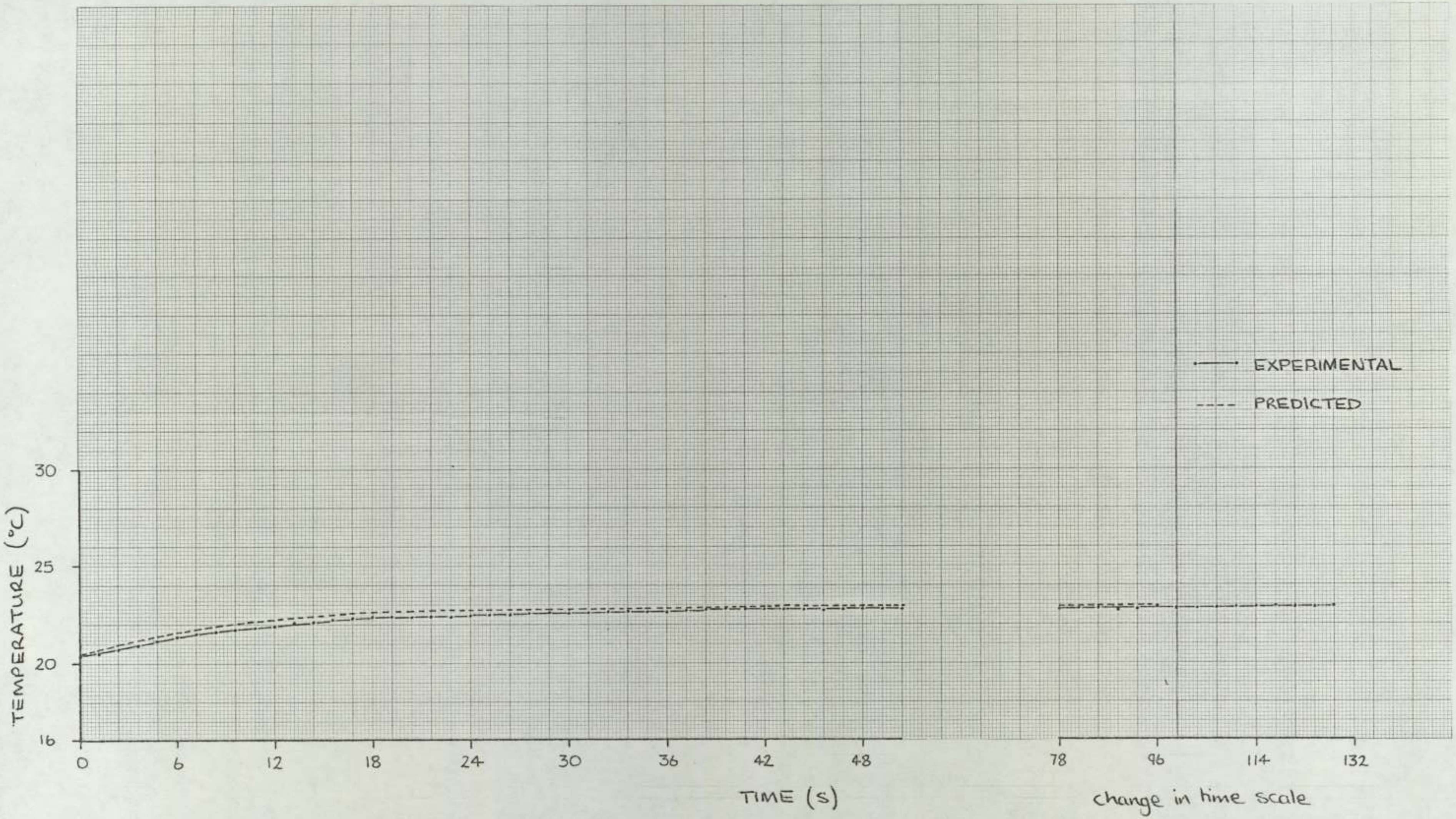


FIG. 5.6 TEMPERATURE TRANSIENT RESPONSE, RUN 7, TUBE OUTLET.

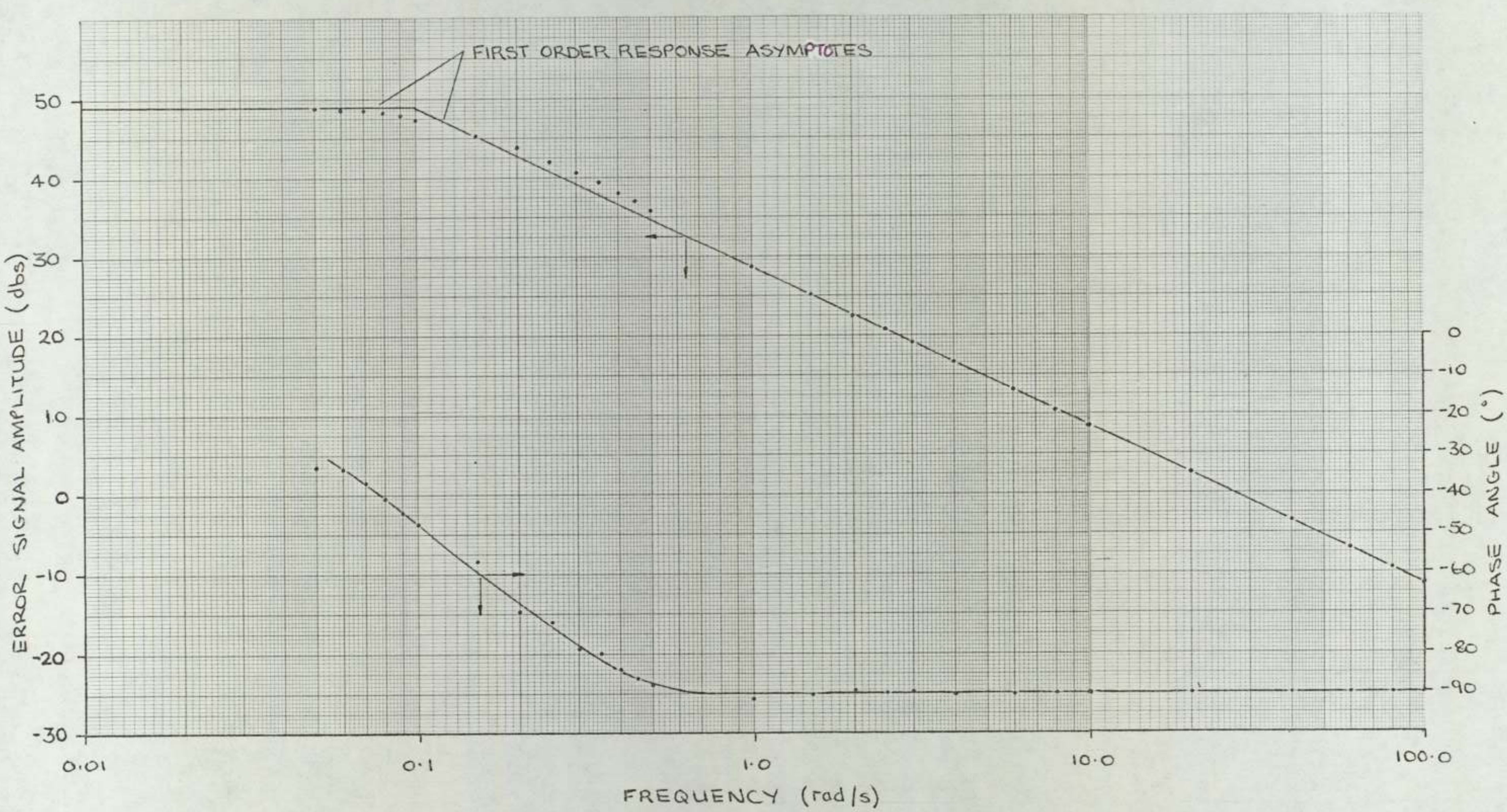


FIG. 5.7 BODE DIAGRAM FOR RUN 1, SHELL INLET TEMPERATURE ERROR SIGNAL.

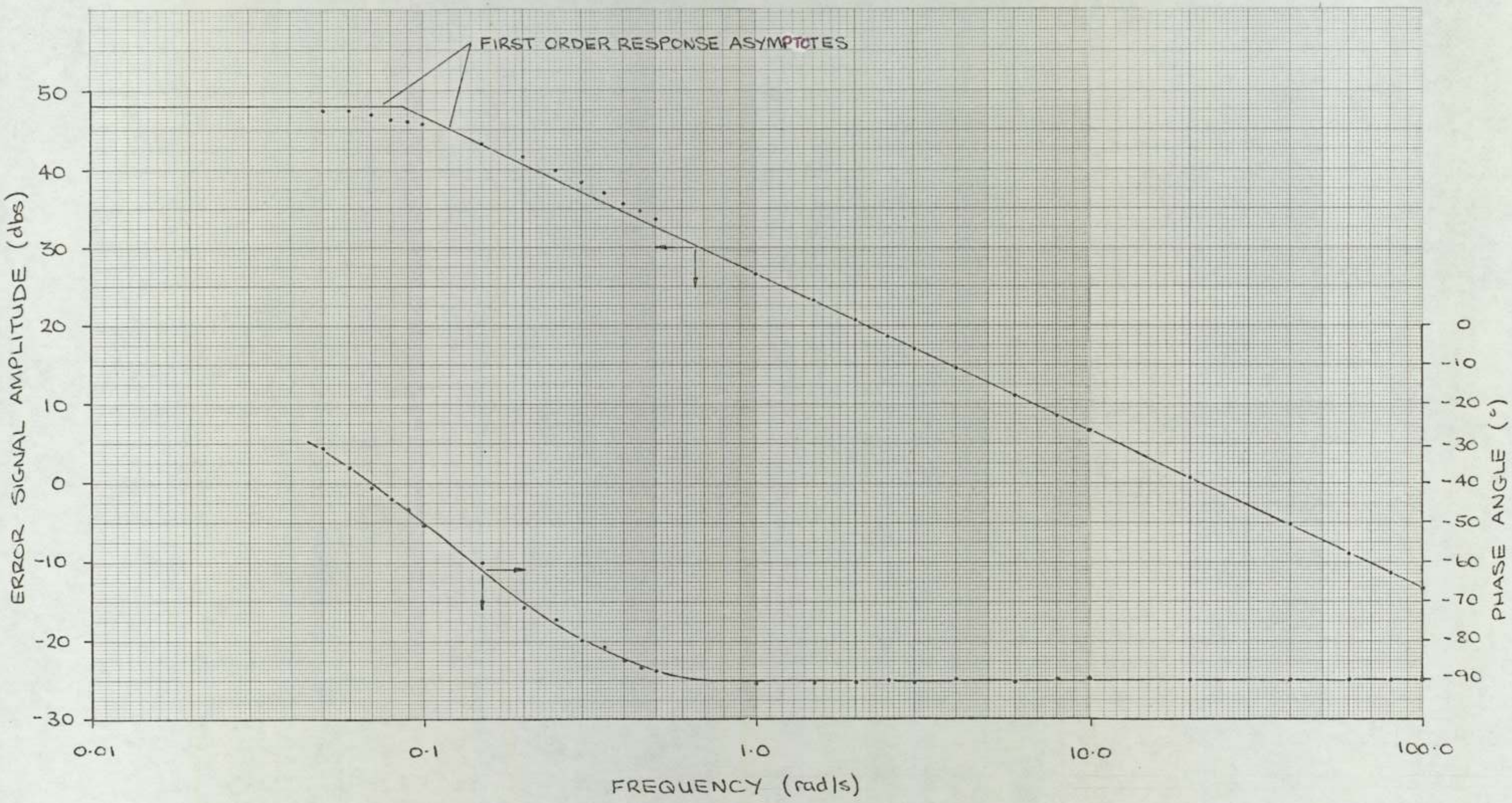


FIG. 5.8 BODE DIAGRAM FOR RUN 1, SHELL OUTLET TEMPERATURE ERROR SIGNAL.

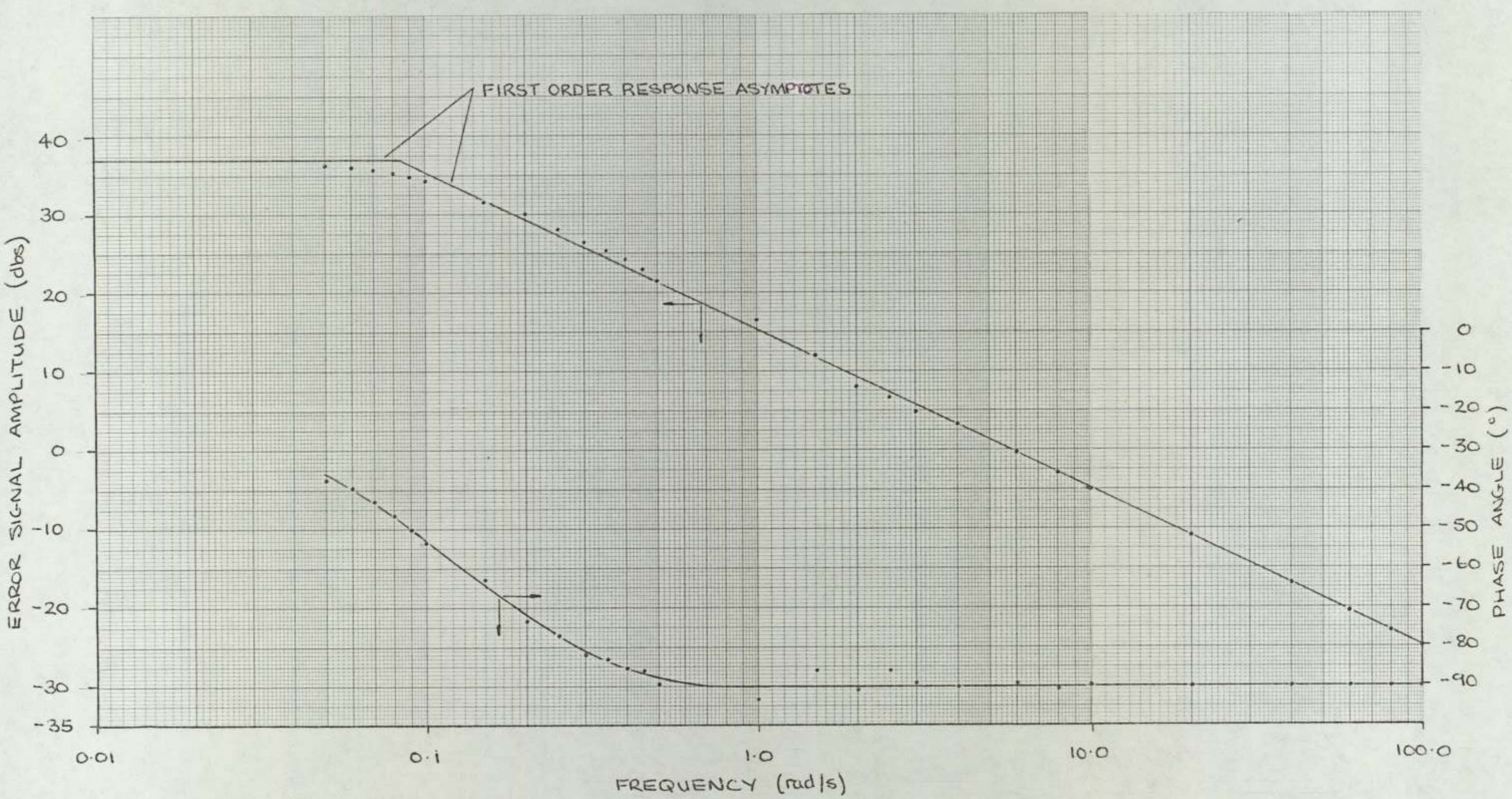


FIG. 5.9 BODE DIAGRAM FOR RUN 1, TUBE OUTLET TEMPERATURE ERROR SIGNAL.

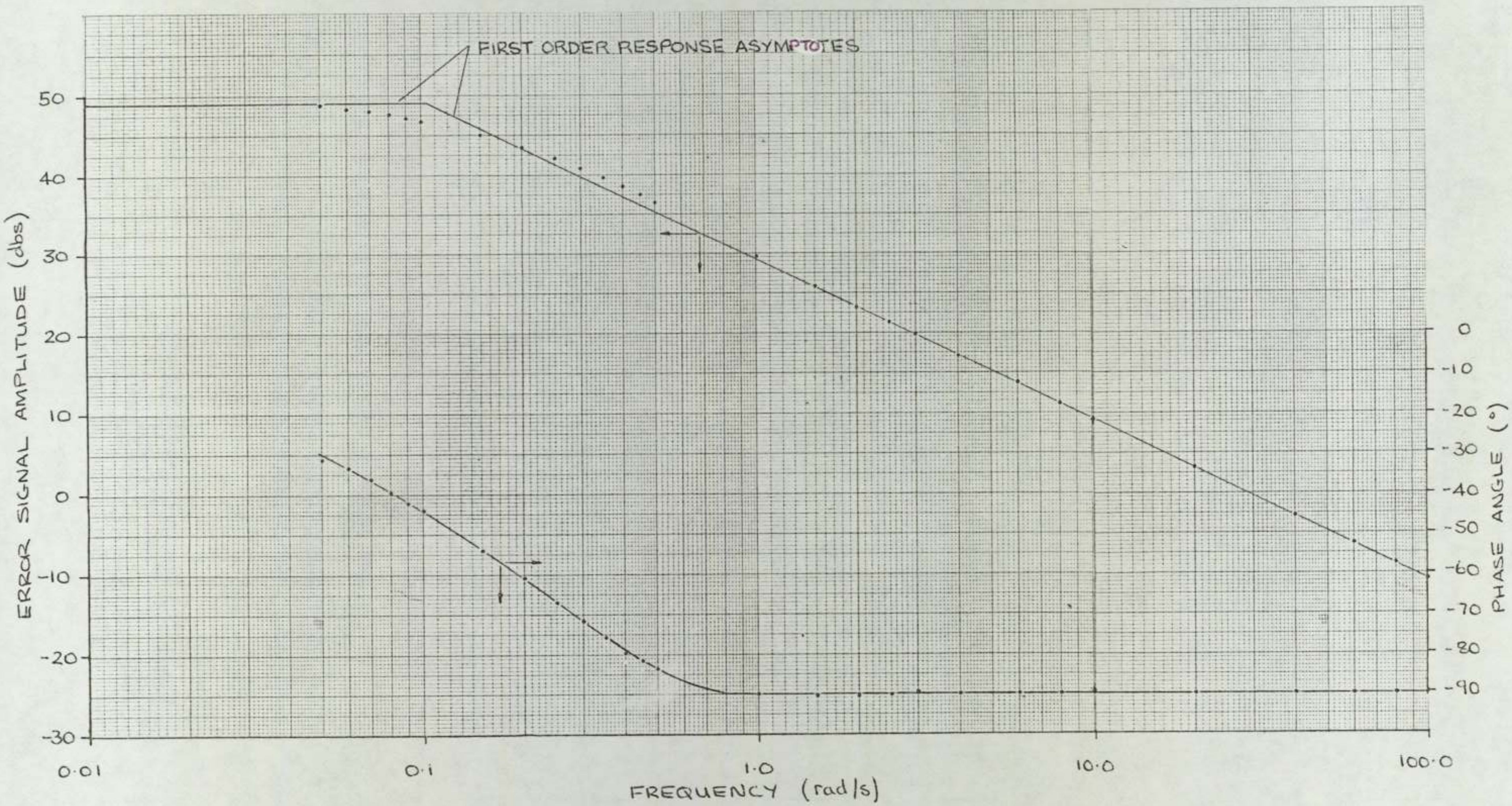


FIG. 5.10 BODE DIAGRAM FOR RUN 7, SHELL INLET TEMPERATURE ERROR SIGNAL

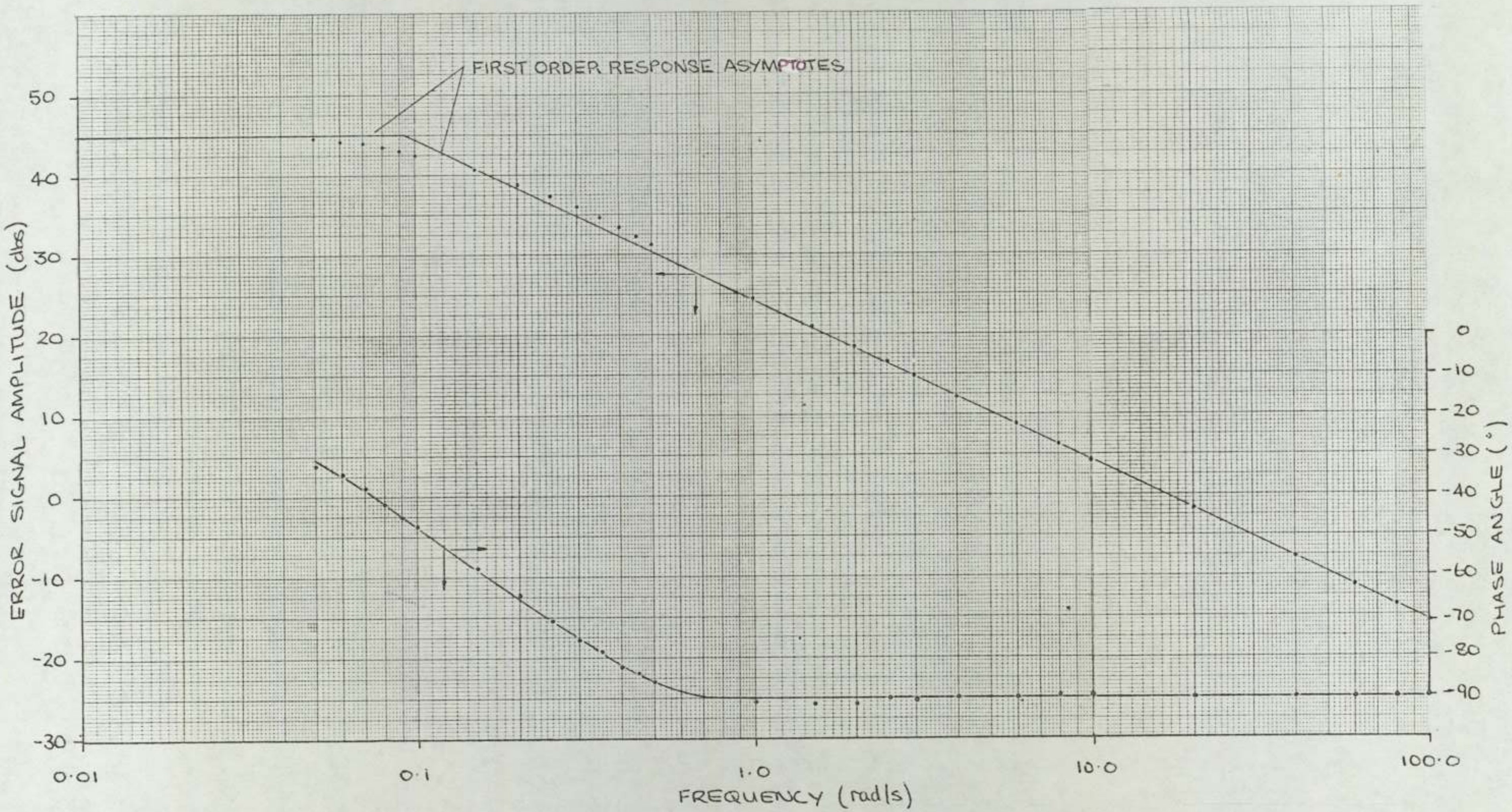


FIG. 5.11 BODE DIAGRAM FOR RUN 7, SHELL OUTLET TEMPERATURE ERROR SIGNAL.

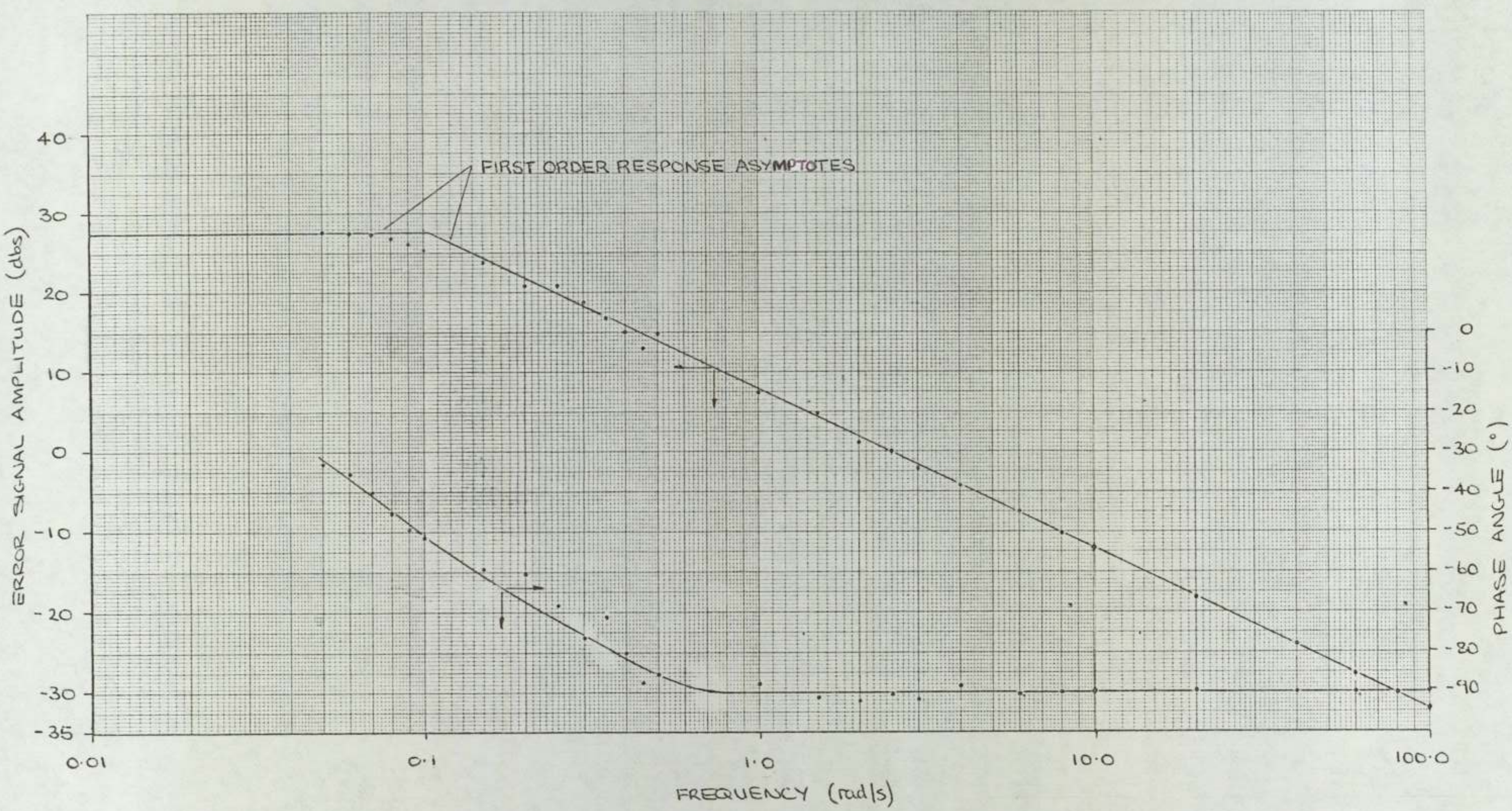
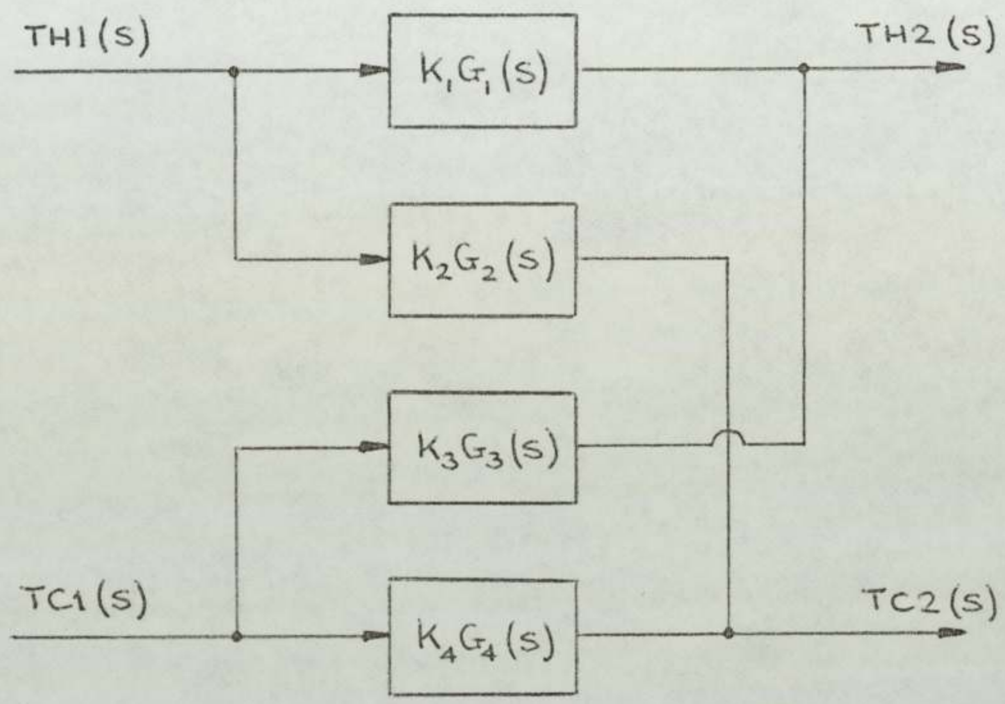


FIG. 5.12 BODE DIAGRAM FOR RUN 7, TUBE OUTLET TEMPERATURE ERROR SIGNAL



where $$\frac{TH2(s)}{TH1(s)} = k_1 G_1(s)$$

$$\frac{TC2(s)}{TH1(s)} = k_2 G_2(s) \qquad \text{etc}$$

FIG. 5.13 SIMPLIFIED MODEL OF HEAT EXCHANGER.

CONCLUSIONS

The work did not progress to the stage where the principle of dynamic testing to determine the thermal performance characteristics of heat exchangers made by Serck could be assessed.

The time available permitted only a limited analysis of the dynamic test results from which it was clear that there were analysis difficulties associated with the experimental data collected and one of the programmes, DFTR. These problems need to be resolved before an indepth analysis can be attempted.

The water/water dynamic tests showed that following the applied temperature disturbance the transient effects had disappeared after approximately 3 minutes. In the oil/water tests, for similar temperature disturbances, a steady state condition was reached after 6 - 9 minutes depending on the flow rate. The short time to reach a steady state is aided by the fact that the dynamic tests utilised a constant temperature source. The relatively short time required to reach a steady state indicates that there is no need to develop a dynamic testing technique to predict the steady state behaviour of a heat exchanger from part of the transient response unless the response is very slow. The work does indicate that by temperature forcing a steady state could be reached quite quickly, providing there is sufficient heating capacity available. However, it is probable that the degree of forcing will not be limited solely by the heat exchanger test unit dynamics but by the dynamics of the other components in the test system.

The principle of dynamic testing has yet to be proved, and if shown to be an adequate testing method would require considerable work to develop the principle into a practical form of testing. Based on these and the above arguments, dynamic testing appears unattractive.

An alternative method of testing would be to consider closed loop control of the heat exchanger to stabilize steady state levels more quickly. Appendix A6 is a report giving a preliminary assessment of automating the steady state testing of heat exchangers.

An alternative definition of steady state which may be worthy of investigation is the determination of the asymptotic value of the instantaneous overall heat transfer coefficient as the coefficient changes with time during the transient. This differs from the usual definition - zero change in temperature with time - in that the heat exchanger inlet and outlet temperatures may vary with time.

Since the initiation of the project 4 years have passed. In the last 18 months the management structure at Serck Heat Transfer has undergone considerable change aimed at preparing the company for business in the 1980's/1990's. Consequently the priorities for new products and hence testing may have changed. Therefore, the work presented in this report and any extention of its needs to be viewed in relation to the company's future business strategy.

RECOMMENDATIONS

The recommendations presented here are two-fold. Firstly, the academic side of the work is considered and, secondly, recommendations based from an industrial view point given.

ACADEMIC

1. Resolve the analysis difficulties. This may in part be achieved by:
   a) Interpolating between existing experimental data points to provide extra data and thereby reducing the sample interval.
   b) A detailed investigation of the numerical accuracy of the Direct Fourier Transform Routine, programme DFTR, at low frequencies, <0.1 rad/s.
2. Develop a model to improve the prediction of the time domain transient response for water to water heat exchange. Similarly, develop a model for the oil/water case.
3. Apply the models to the prediction of the steady state response of the heat exchanger using the response curves to assess the suitability of dynamic testing as a technique for predicting the thermal performance of heat exchangers, or the performance of similar processes. This technique could be of value where the process responds very slowly.
4. Investigate the correlation of the model time constants and gains to the physical parameters of the heat exchanger and fluids to yield models of relatively simple form for use in the analysis of heat exchanger dynamics.

INDUSTRIAL

1. The company note that the investigation into the thermal performance testing of heat exchangers using the dynamic testing technique described in this report gives no particular advantage and would require considerable work to develop it into a practical form of testing. This approach should, therefore, be discontinued.
2. The company to use the report given as Appendix A6, titled, "Automating the Steady State Testing of Heat Exchangers - a Preliminary Assessment," as an initial basis for considering automation as a viable testing technique for the future.
3. Investigate the possibility of utilising the concept of the asymptotic value of the overall heat transfer coefficient in determining the steady state behaviour of a heat exchanger.
4. Incorporate recommendation 3 and the concept of forcing the test unit to a steady state condition in the design of an automated test facility.

APPENDIX A1

THE HEAT EXCHANGER TEST UNIT

APPENDIX A1

THE HEAT EXCHANGER TEST UNIT

GENERAL

Manufacturer:	Serck Heat Transfer
Designation:	AA39 - B KEF7
General Arrangement drawing no:	45511-4101

DESCRIPTION

Type:	Shell & Tube
Flow arrangement:	1 tubeside fluid pass
	1 shellside fluid pass
Baffle arrangement:	Disc & Doughnut,
	4 doughnut baffles
	3 disc baffles
Tube layout:	on circular pitch
Number of tubes:	60
Tube size:	nominal outside dia. 6.25 mm
	wall thickness 0.35 mm
L/D ratio:	30
Heat transfer surface area:	0.18063 m^2 (nominal)
Materials: Tubestack:	Aluminium brass tubes & naval brass tube plates
Cylinder:	Aluminium
Water boxes:	Gun metal
Mass of unit (empty):	
Cylinder:	0.829 kg
Tubestack:	0.815 kg
2 Water boxes:	1.098 kg
6 bolts:	0.050 kg
Total	2.792 kg

The general arrangement drawing is shown in fig. Al.1 and the tubestack drawing in fig. Al.2.

The test unit is shown in Photograph 1.

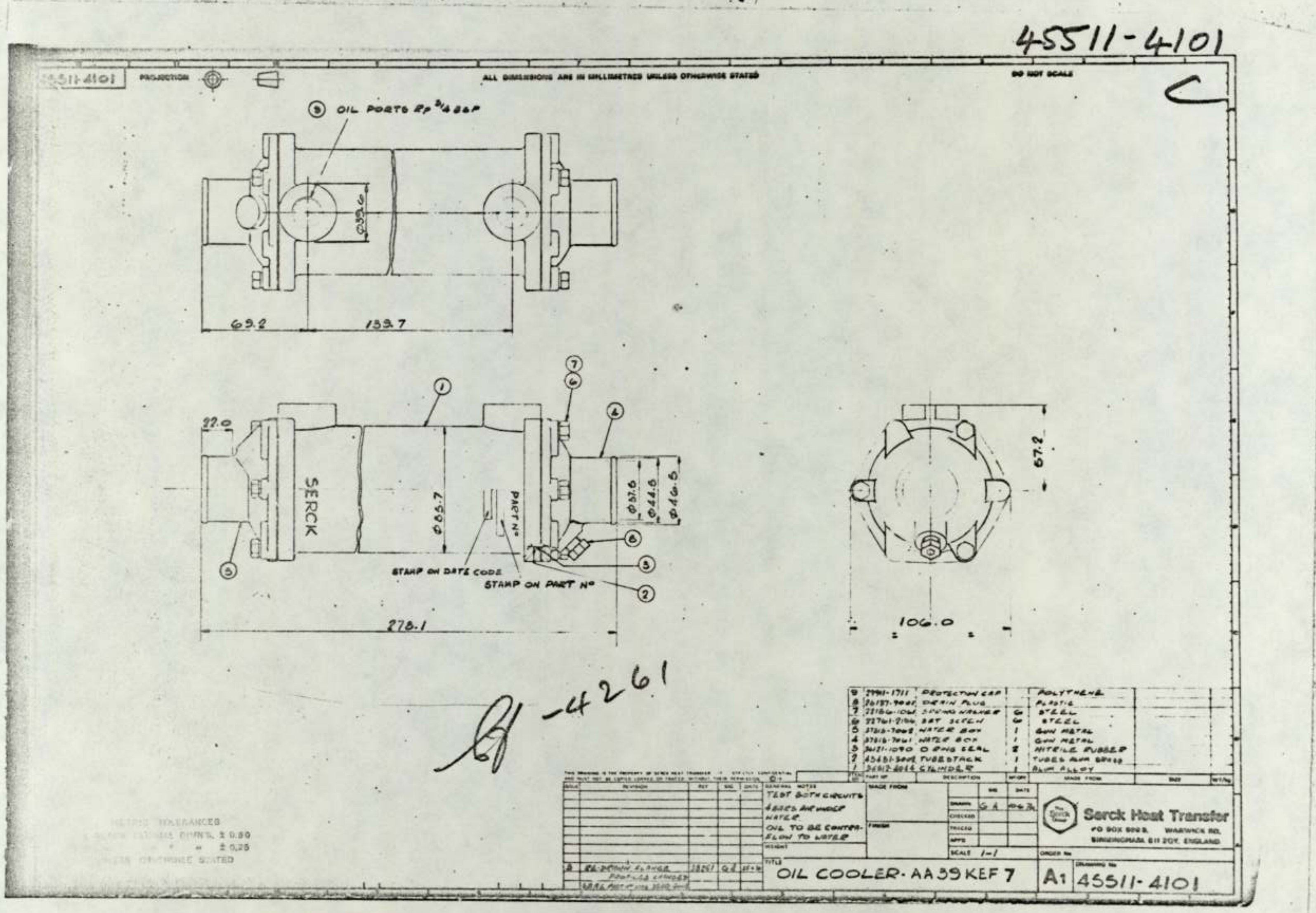


FIG.A1.1 GENERAL ARRANGEMENT DRAWING OF TEST UNIT

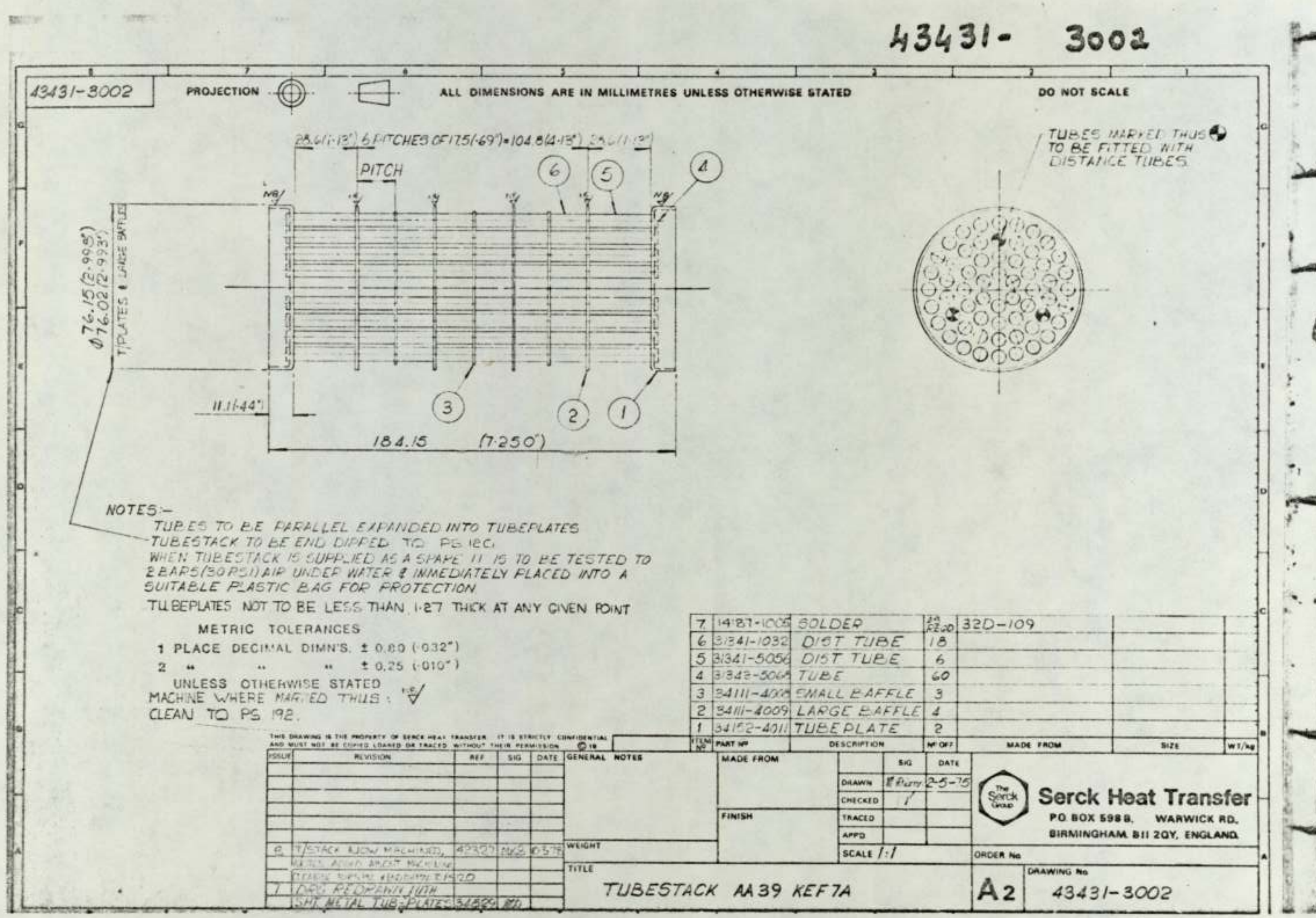
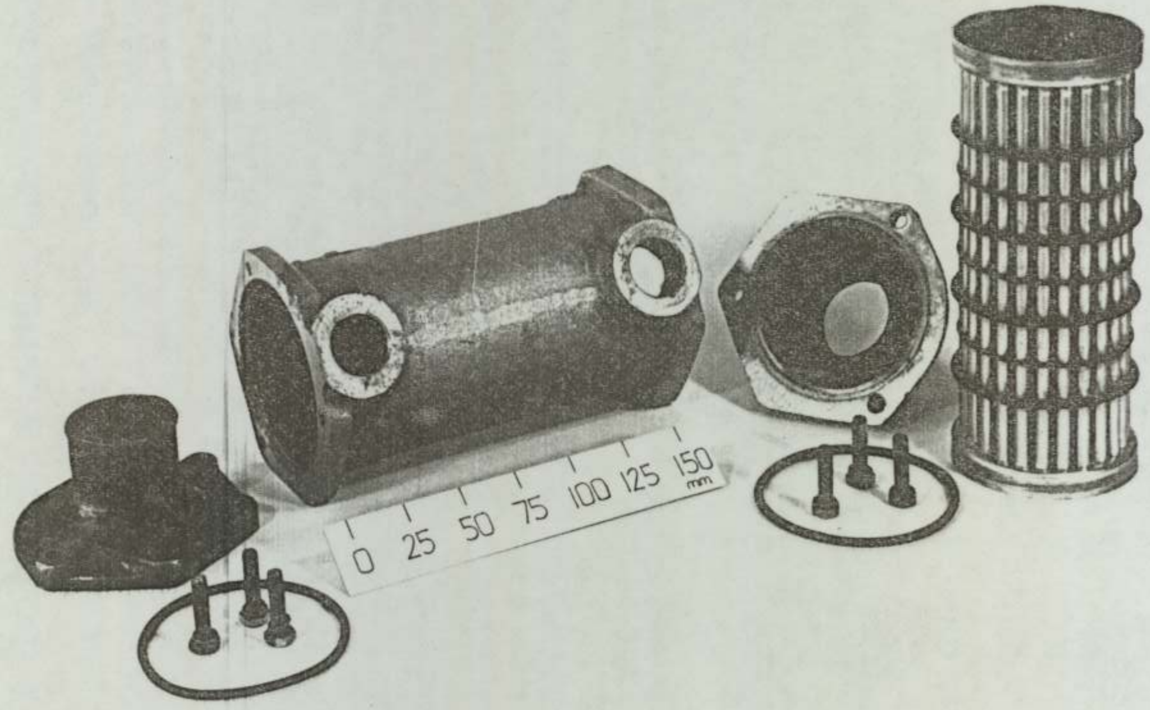


FIG.A1.2. TUBESTACK DRAWING FOR TEST UNIT



PHOTOGRAPH 1. THE HEAT EXCHANGER TEST UNIT.

APPENDIX A2

DERIVATION OF NUMERICAL ROUTINE AND PROGRAMME LISTINGS FOR DYNAMIC TEST METHOD

A2.1. DERIVATION OF THE DIRECT FOURIER TRANSFORM NUMERICAL PROCEDURE

The frequency response characteristics $G(\omega)$ of a time function $f(t)$ can be determined using the Direct Fourier Transform:

$$G(\omega) = \int_{-\infty}^{+\infty} f(t)\, e^{-j\omega t} dt \qquad \text{(A2.1)}$$

When $f(t)$ is not given analytically, $G(\omega)$ can be computed numerically.

Consider a segment of the transient response $f(t)$, fig. A2.1, and assume a linear relationship between t_n and t_{n+1}.

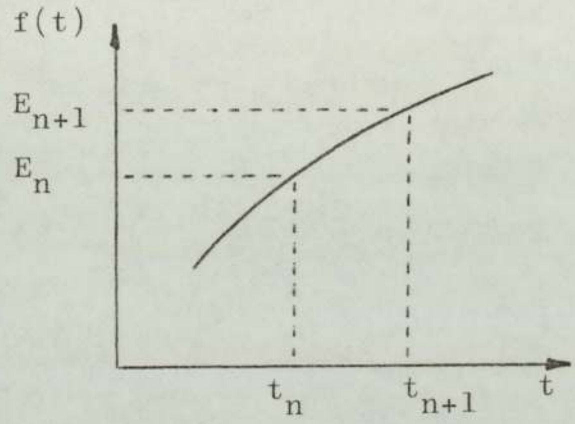


Fig A2.1

$$f(t) = m_n t + c_n \qquad \text{(A2.2)}$$

$$\text{where } m_n = \frac{E_{n+1} - E_n}{t_{n+1} - t_n}$$

$$c_n = f(t) - m_n t = E_n - \left[\frac{E_{n+1} - E_n}{t_{n+1} - t_n}\right] t$$

The Direct Fourier Transform over the interval t_n to t_{n+1} is obtained by substituting equation A2.2 into A2.1, thus:

$$G_n(\omega) = m_n \int_{t_n}^{\infty} te^{-j\omega t} dt + c_n \int_{t_n}^{\infty} e^{-j\omega t} dt - m_n \int_{t_{n+1}}^{\infty} te^{-j\omega t} dt$$

$$-c_n \int_{t_{n+1}}^{\infty} e^{-j\omega t} dt \qquad \text{(A2.3)}$$

and integrating by parts and simplifying to give:

$$G_n(\omega) = -\frac{1}{\omega^2}\left[\frac{E_{n+1} - E_n}{t_{n+1} - t_n}\right]\left(e^{-j\omega t_n} - e^{-j\omega t_{n+1}}\right)$$

$$+\frac{1}{j\omega}\left(E_n e^{-j\omega t_n} - E_{n+1}e^{-j\omega t_{n+1}}\right) \qquad \text{(A2.4)}$$

The Direct Fourier Transform for the complete transient is given by:

$$G(\omega) = \sum G_n(\omega) \qquad \text{(A2.5)}$$

which when $t_o = 0$ and $f(t_k) = 0$ becomes:

$$G(\omega) = \frac{E_o}{j\omega} + \sum_{n=0}^{k-1} -\frac{1}{\omega^2}\left[\frac{E_{n+1} - E_n}{t_{n+1} - t_n}\right](e^{-j\omega t_n} - e^{-j\omega t_{n+1}})$$

(A2.6)

Remembering $e^{j\omega t} = \cos\omega t + j\sin\omega t$

and $e^{-j\omega t} = \cos\omega t - j\sin\omega t$

$$G(\omega) = \frac{-jE_o}{\omega} - \frac{1}{\omega^2}\sum_{n=0}^{k-1}\left[\frac{E_{n+1} - E_n}{t_{n+1} - t_n}\right]\Big\{(\cos\omega t_n - \cos\omega t_{n+1})$$

$$-j(\sin\omega t_n - \sin\omega t_{n+1})\Big\} \qquad \text{(A2.7)}$$

Equation A2.7 can be evaluated numerically more efficiently when expressed as:

$$G(\omega) = \frac{-jE_o}{\omega} + \frac{(E_1 - E_o)}{-\Delta t\,\omega^2} + \sum_{n=1}^{k-1}(E_{n+1} - 2E_n + E_{n-1})\Big\{$$

$$\frac{\cos(-\omega t_n) + j\sin(-\omega t_n)}{-\Delta t\,\omega^2}\Big\} \qquad \text{(A2.8)}$$

where $\Delta t = t_{n+1} - t_n$

-1-

```
C PROGRAM SORT, SORTS DYNAMIC TEST DATA & OUTPUTS TRANSIENT
C RESPONSE DATA ON TO PAPER TAPE FOR SUBSEQUENT EVALUATION
C OF THE FREQUENCY RESPONSE USING PROGRAM DFTR.
C LANGUAGE-F77
      DIMENSION L(300,4),C(?,?),E(?,?)
      INTEGER P,Q,R,T,U,Y,G,E1(4),B1,V(12)
      REAL LAG,LAGS,LAGT,M1,M2,MAX,MIN
      STRING B
      PRINT,"WATER/WATER DATA INPUT 1;OIL/WATER DATA INPUT 2 "
      INPUT,B1
      PRINT,"INPUT FILENAME "
      INPUT,B
      PRINT,"INPUT SHELL FLOW,LB/HR "
      INPUT,SF
      PRINT,"INPUT TIME INTERVAL,NO. OF ROWS IN INCREMENT "
      INPUT,T
      PRINT,"+VE/-VE DATA: +VE STEP ONLY,INPUT 1;+VE  & -VE STEP,
     &INPUT 2;"
      PRINT,"-VE STEP ONLY,INPUT 3; +VE DATA ONLY: INPUT 4 "
      INPUT,G
      PRINT,"IF START OF STEP OUTPUT REQUIRED INPUT 1,IF NOT 2 "
      INPUT,Z
      DO 10 I=1,300
      READ (FILE=B,FMT=20,END=30)(V(J),J=1,12)
      L(I,1)=V(2)
      L(I,2)=V(5)
      L(I,3)=V(8)
      L(I,4)=V(11)
   10 CONTINUE
   20 FORMAT(4(I3,I6,I2,2X))
   30 K=I-2
      REDIMENSION C(K,4)
      K2=0
      DO 40 I=1,K
      D=0.001
      V1=L(I,1)*D
      V2=L(I,2)*D
      V3=L(I,3)*D
      V4=L(I,4)*D
      C(I,1)=1.631689E-2+6.484883*V1-6.534999E-2*V1**2+
     &3.822533E-3*V1**3-1.485299E-4*V1**4
      C(I,2)=-2.144337E-3+6.509252*V2-7.196949E-2*V2**2+
     &4.328189E-3*V2**3-1.588805E-4*V2**4
      C(I,3)=2.467203E-2+6.478119*V3-6.245087E-2*V3**2+
     &3.456007E-3*V3**3-1.329752E-4*V3**4
      C(I,4)=3.481293E-2+6.453876*V4-5.432990E-2*V4**2+
     &2.452989E-3*V4**3-9.468164E-5*V4**4
   40 CONTINUE
      IJ=1
      X=0.04
```

-2-

```
      X1=0.2
      X2=0.2
   50 DO 60 I=IJ,K
      IF(I.EQ.1)M1=ABS(C(I,1)-C((I+1),1))
      IF(I.GT.1)M1=ABS(C((I-1),1)-C(I,1))
      M2=ABS(C(I,1)-C((I+1),1))
      IF(M1.GE.X.AND.M2.GE.X)GOTO 70
   60 CONTINUE
   70 K1=I
      IF(G.EQ.4)GOTO 100
      P=K1+100
      DO 80 I=P,K
      M1=ABS(C((I-1),1)-C(I,1))
      M2=ABS(C(I,1)-C((I+1),1))
      IF(M1.GE.X.AND.M2.GE.X)GOTO 90
   80 CONTINUE
   90 K2=I
  100 IF(Z.GT.1.5)GOTO 150
      P=K1-10
      Q=K1+10
      IF(P.LE.0)P=1
      DO 110 I=P,Q
      IF(I.EQ.K1)PRINT,'*****'
      PRINT 130,I,(C(I,J),J=1,4)
      IF(I.EQ.K1)PRINT,'*****'
  110 CONTINUE
      PRINT 140
      IF(G.EQ.4)GOTO 150
      P=K2-15
      Q=K2+10
      DO 120 I=P,Q
      PRINT 130,I,(C(I,J),J=1,4)
  120 CONTINUE
  130 FORMAT(1X,I3,5X,4(F5.2,5X))
  140 FORMAT(1H )
  150 IF(B1.EQ.1)THEN
      LAG=10573.0/SF
      S1=0.3
      S2=0.9
      S3=1.2
      ELSE
      LAG=14924.0/SF
      S1=1.4
      S2=4.2
      S3=5.6
      END IF
      K3=K2-1
      IF(G.EQ.4)K3=K
      IF(G.EQ.3)GOTO 430
  160 K9=K1
```

A2.2.1 PROGRAMME 'SORT'

-3-

```
      K10=K1
      PRINT 170,X,X1,X2
  170 FORMAT(1X,'TEMP DIFF PARAMETERS ARE: ',2(F4.2,1H,),F4.2)
      PRINT,"NEW TEMP DIFF PARAMETERS REQUIRED? NO 1; YES 2 "
      INPUT,JJ
      IF(JJ.EQ.1)GOTO 180
      PRINT,"RE-LOCATE START OF SHELL INLET RESPONSE? NO 1;
     & YES 2 "
      INPUT,JK
      IF(JK.EQ.2)THEN
      PRINT,"NEW SHELL INLET TEMP DIFF= "
      INPUT,X
      PRINT,"START SEARCH AT ROW NO: "
      INPUT,IJ
      GOTO 50
      END IF
      PRINT,"NEW SHELL OUTLET TEMP DIFF= "
      INPUT,X1
      PRINT,"SHELL OUTLET RESPONSE INDICATED IN ROW: "
      INPUT,K9
      PRINT,"NEW TUBE OUTLET TEMP DIFF= "
      INPUT,X2
      PRINT,"TUBE OUTLET RESPONSE INDICATED IN ROW: "
      INPUT,K10
  180 CONTINUE
      DO 190 I=K9,K3
      M1=ABS(C((I-1),2)-C(I,2))
      M2=ABS(C(I,2)-C((I+1),2))
      IF(M1.GE.X1.AND.M2.GE.X1)GOTO 200
  190 CONTINUE
  200 K4=I
      DO 210 I=K10,K3
      M1=ABS(C((I-1),4)-C(I,4))
      M2=ABS(C(I,4)-C((I+1),4))
      IF(M1.GE.X2.AND.M2.GE.X2)GOTO 220
  210 CONTINUE
  220 K5=I
      LAGS=S3*(K4-K1)+S1
      LAGT=S3*(K5-K1)+S2
      K7=K3-LAG/S3-1
      IF(K3.EQ.K)K7=K
      N1=(K7-(K1-1))/T+1
      REDIMENSION E(N1,8)
      M=K1
      R=0
      DO 280 J=2,8,2
      IF(K3.EQ.K)K7=K
      K6=((K7-(M-1)+1)/T)*T+(M-1)
      IF(T.EQ.1)K6=K7
      R=R+1
```

-4-

```
      E1(R)=(K6-(M-1))/T+1
      N=1
      E(N,J)=C((M-1),R)
      TIME=0
      E(N,(J-1))=TIME
      Y=M+T-1
      DO 230 I=Y,K6,T
      N=N+1
      E(N,J)=C(I,R)
      TIME=TIME+S3*T
      E(N,(J-1))=TIME
  230 CONTINUE
      GOTO(240,250,260,270)R
  240 M=K4
      GOTO 270
  250 M=K1
      GOTO 270
  260 M=K5
  270 CONTINUE
  280 CONTINUE
      MIN=10.0**6
      MAX=-10.0**6
      SUM1=0.0
      SUM2=0.0
      K8=K1-LAG/S3-1
      DO 290 I=K8,K7
      IF(C(I,3).LE.MIN)MIN=C(I,3)
      IF(C(I,3).GE.MAX)MAX=C(I,3)
      SUM1=SUM1+C(I,3)
      SUM2=SUM2+C(I,3)**2
  290 CONTINUE
      SN=K7-K8+1
      AV=SUM1/SN
      SD=SQRT((SUM2-(SUM1**2)/SN)/(SN-1))
      PRINT,'DYNAMIC TEST RUN NO.-',B
      PRINT 300,LAG
  300 FORMAT (1X,46HDELAY BEFORE TEMP. STEP ARRIVES AT SHELL
     & INLET,2X,F5.2,1X,3HSEC)
      PRINT 310,LAGS
  310 FORMAT (1X,16HSHELL OUTLET LAG,2X,F5.2,1X,3HSEC)
      PRINT 320,LAGT
  320 FORMAT (1X,15HTUBE OUTLET LAG,2X,F5.2,1X,3HSEC)
      PRINT 140
      PRINT,"COPY OF TIME/TEMP DATA REQUIRED? ALL 1; PART 2;
     & NONE 3"
      INPUT,C1
      IF(C1.GT.2.5)GOTO 350
      N3=N1
      IF(C1.GT.1.5)N3=10
      PRINT,'TIME IN SEC;  TEMP IN DEG C'
```

```
      PRINT,' TIME     TH1      TIME     TH2      TIME     TC1
     &TIME     TC2'
      DO 340  I=1,N3
      PRINT 330,(E(I,J),J=1,8)
  330 FORMAT (1X,4(F6.1,2X,F5.2,3X))
  340 CONTINUE
  350 PRINT 140
      PRINT,'INITAL TEMPERATURES; DEG C.'
      PRINT 360
  360 FORMAT (1X,25H TH1     TH2     TC1     TC2)
      PRINT 370,(E(1,J),J=2,8,2)
  370 FORMAT (1X,4(F5.2,2X))
      L1=E1(1)
      L2=E1(2)
      L3=E1(3)
      L4=E1(4)
      PRINT 140
      PRINT,'FINAL TEMPERATURES; DEG C.'
      PRINT 360
      PRINT 370,E(L1,2),E(L2,4),E(L3,6),E(L4,8)
      PRINT 140
      STEP=E(L1,2)-E(1,2)
      PRINT 380,STEP
  380 FORMAT(1X,'MAGNITUDE OF STEP INPUT=  ',F5.2,' DEG C')
      PRINT 140
      PRINT 390,AV
  390 FORMAT(1X,'AVERAGE TUBE INLET TEMPERATURE= ',F5.2,
     &'  DEG C')
      PRINT 400,MIN,MAX
  400 FORMAT(1X,'MIN= ',F5.2,'  MAX= ',F5.2)
      PRINT,'STANDARD DEVIATION OF TUBE INLET TEMPERATURE= ',SD
      PRINT 140
      PRINT,'NO. OF TEMPERATURE SAMPLES:'
      PRINT,'  TH1=',L1
      PRINT,'  TH2=',L2
      PRINT,'  TC1=',L3
      PRINT,'  TC2=',L4
      J2=0
      DO 420 J=2,8,2
      J2=J2+1
      N2=E1(J2)
      PRINT,''
      PRINT 410,(E(I,J),I=1,N2)
  410 FORMAT(1X,F5.2)
      PRINT,''
      PAUSE
      PAUSE 'PAUSED,FEED BLANK TAPE'
  420 CONTINUE
      IF(C(K3,1).LT.40.0)GOTO 440
      IF(G.EQ.1.OR.G.EQ.4)GOTO 440
```



```
  430 K1=K2
      K3=K
      REDIMENSION E(0,0)
      GOTO 160
  440 STOP
      END
```

A2.2.2 PROGRAMME 'DFTR'

```
C   PROGRAM:DFTR
C   THIS PROGRAM IS A DIRECT FOURIER TRANSFORM ROUTINE
C   WHICH EVALUATES THE FREQUENCY RESPONSE FROM THE
C   TRANSIENT RESPONSE.
C   THE INPUT TO DFTR IS THE OUTPUT FROM PROGRAM SORT.
C   THE OUTPUT FROM DFTR IS THE INPUT FOR PROGRAM FITFRQ.
C
C   LANGUAGE.....FORTRAN IV
C
C   UNIT NUMBERS ASSIGNED TO:
C   1........USER TELETYPE FOR READING OF INPUT PARAMETERS
C            REQUESTED BY PROGRAM
C   2........VDU DISPLAY
C   9........OUTPUT FILE: SHELLINLET FREQUENCY RESPONSE
C   10.......OUTPUT FILE: SHELLOUTLET FREQUENCY RESPONSE
C   11.......OUTPUT FILE: TUBEOUTLET FREQUENCY RESPONSE
C   13.......INPUT FILE: SHELLINLET TRANSIENT RESPONSE
C   14.......INPUT FILE: SHELLOUTLET TRANSIENT RESPONSE
C   15.......INPUT FILE: TUBEOUTLET TRANSIENT RESPONSE
C   16.......ERROR TRACING FILES
C   UNITS 9-11,13-16 ARE FILES ON THE 'SCRATCH DISC'
C
      DIMENSION A(200,3),Z(200),AO(3)
      WRITE(16,98)
   98 FORMAT(1H ,'ENTER DFTR')
      READ(13,100,END=10)(A(I,1),I=1,200)
   10 K1=I-1
      AO(1)=A(K1,1)-A(1,1)
      DO 20 I=2,K1
   20 A(I-1,1)=A(K1,1)-A(I,1)
      READ(14,100,END=11)(A(I,2),I=1,200)
   11 K2=I-1
      AO(2)=A(K2,2)-A(1,2)
      DO 21 I=2,K2
   21 A(I-1,2)=A(K2,2)-A(I,2)
      READ(15,100,END=12)(A(I,3),I=1,200)
   12 K3=I-1
      AO(3)=A(K3,3)-A(1,3)
      DO 22 I=2,K3
   22 A(I-1,3)=A(K3,3)-A(I,3)
  100 FORMAT(F5.2)
      WRITE(16,109)K1,K2,K3
  109 FORMAT('K1= ',I4,'  K2= ',I4,'  K3= ',I4)
      DO 25 I=1,5
   25 WRITE(16,101)(A(I,J),J=1,3)
      WRITE(16,101)A(K1-2,1),A(K2-2,2),A(K3-2,3)
      WRITE(16,101)A(K1-1,1),A(K2-1,2),A(K3-1,3)
  101 FORMAT(3(F5.2,5X))
      WRITE(2,102)
  102 FORMAT(1H ,'SAMPLE INTERVAL: 1.2S =1, 2.0S =2, 5.6S =3 ?...'/)
      READ(1,103)L1
      WRITE(2,111)
  111 FORMAT(1H ,'NO. OF RESPONSES: 1,2 OR 3 ?...')
      READ(1,103)K
  103 FORMAT(I1)
      D=1.2
      IF(L1.EQ.2)D=2.0
      IF(L1.EQ.3)D=5.6
      WRITE(16,112)D
  112 FORMAT('SAMPLE INTERVAL= ',F3.1,' SEC')
   31 WRITE(2,104)
  104 FORMAT(1H ,'START FREQ=?...')
```

```
      READ(1,105)W1
      WRITE(2,106)
  106 FORMAT(1H ,'END FREQ=?...')
      READ(1,105)W2
      WRITE(2,107)
  107 FORMAT(1H ,'FREQ STEP=?...')
      READ(1,105)W3
  105 FORMAT(F8.4)
      DO 30 I1=1,K
      IF(I1.EQ.1)L=K1-2
      IF(I1.EQ.2)L=K2-2
      IF(I1.EQ.3)L=K3-2
      W=W1
      I=0
    1 IF(W.GT.W2)GOTO 32
      I=I+1
      DIV=-D*W*W
      T=0.0
      SUMRE=(A(1,I1)-A0(I1))/DIV
      SUMXI=-A0(I1)/W
      DO 2 J=1,L
      T=T+D
      J1=J+1
      JM1=J-1
      IF(I1-1)9,9,8
    9 IF(J-1)3,3,7
    3 Z(1)=A(J1,I1)-2.0*A(J,I1)+A0(I1)
      GOTO 8
    7 Z(J)=A(J1,I1)-2.0*A(J,I1)+A(JM1,I1)
    8 W4=-W*T
      RE=COS(W4)*Z(J)
      XI=SIN(W4)*Z(J)
      SUMRE=SUMRE+RE/DIV
    2 SUMXI=SUMXI+XI/DIV
      XM=20.0*ALOG10(SQRT(SUMRE**2+SUMXI**2))
C     IF VECTOR IN +RE,+IM QUADRANT,PHASE ANGLE IS LEADING:
C     IN OTHER 3 QUADRANTS,PHASE ANGLE IS LAGGING.
      IF(SUMRE)5,6,6
    5 PHD=(ATAN(SUMXI/SUMRE))*180.0/3.1416-180.0
      GOTO 15
    6 PHD=(ATAN(SUMXI/SUMRE))*180.0/3.1416
   15 IF(I1.EQ.1)WRITE(9,108)W,XM,PHD
      IF(I1.EQ.2)WRITE(10,108)W,XM,PHD
      IF(I1.EQ.3)WRITE(11,108)W,XM,PHD
  108 FORMAT(3(1X,F12.5))
      W=W+W3
      GOTO 1
   32 CONTINUE
   30 CONTINUE
      WRITE(2,110)
  110 FORMAT(1H ,'CONTINUE FREQ EVALUATION: NO=0, YES=1 ?...')
      READ(1,103)L2
      IF(L2.EQ.1)GOTO 31
      CONTINUE
      WRITE(16,99)
   99 FORMAT(1H ,'EXIT DFTR')
      STOP
      END
```

A2.2.3 MASTER PROGRAMME 'FITFRQ' AND SUBROUTINE 'GAIN ' ONLY

```
C    PROGRAM:FITFRQ
C    MASTER PROGRAM FOR FITTING COMPLEX COEFFICIENTS OF THE
C    DIRECT FOURIER TRANSFORM TO A RATIO OF COMPLEX POLYNOMIALS
C    IN THE FREQUENCY DOMAIN.
C    THE INPUT TO FITFRQ IS THE OUTPUT FROM PROGRAM DFTR.
C    THE OUTPUT FROM FITFRQ IS THE INPUT FOR PROGRAM ESTOCS.
C
C    FITFRQ CALLS THE FOLLOWING ROUTINES:
C    FFIT.........A CURVE FITTING ROUTINE
C    MATINV.......A MATRIX INVERSION ROUTINE(CALLED BY FFIT)
C    MMLT.........A MATRIX MULTIPLICATION ROUTINE(CALLED BY FFIT)
C    RIMP.........A POLAR TO VECTOR CONVERSION ROUTINE
C    NEWER........A NEWTON-RAPHSON ROUTINE
C    POLVAL2......A POLYNOMIAL COEFFICIENT ROUTINE(CALLED BY NEWER)
C    GAIN2........A ROUTINE USED IN THE CALCULATION OF THE STEADY
C                 STATE GAIN
C
C    LANGUAGE.....FORTRAN IV
C
C    UNIT NUMBERS ASSIGNED TO:
C    1........USER TELETYPE FOR READING OF INPUT PARAMETERS REQUESTED
C             BY PROGRAM
C    2........VDU DISPLAY
C    3........FILE FOR TRACING OPERATION OF PROGRAM & SHOWING RESULTS
C             OF INTERMEDIATE CALCULATIONS
C    4........UNASSIGNED
C    5,6,7....OUTPUT FILES: ASSIGNED AS REQUESTED BY PROGRAM
C    8........UNASSIGNED
C    9,10,11..INPUT FILES: ASSIGNED AS REQUESTED BY PROGRAM
C    12.......OUTPUT FILE FOR SUMMARY OF RESULTS
C    UNITS 3,5-7,9-11,12 ARE FILES ON THE 'SCRATCH DISC'
C
      DIMENSION XX(10)
      INTEGER OUT
      REAL R(50),I(50),F(50),SPN(14),SPD(14)
      J2=0
   14 WRITE(2,19)
   19 FORMAT(23H      TYPE 0 TO BEGIN.../)
      READ(1,251)IF
  251 FORMAT(I1)
      IF(IF-1)22,14,14
   22 WRITE(2,250)
  250 FORMAT(25H    NO.OF DATA POINTS?...)
      READ(1,252)K
  252 FORMAT(I3)
      WRITE(3,700)K
  700 FORMAT(23H    NO.OF DATA POINTS =,I3)
      IF(K.GT.200) GO TO 22
   63 WRITE(2,350)
  350 FORMAT(1H ,'UNIT NUMBER FOR INPUT DATA?...')
      READ(1,351)IN
  351 FORMAT(I2)
      WRITE(3,701)IN
  701 FORMAT(1H ,'UNIT NUMBER FOR INPUT DATA=  ',I2)
      WRITE(2,60)
   60 FORMAT(1H ,'UNIT NUMBER FOR OUTPUT DATA?...')
      READ(1,351)OUT
      WRITE(3,62)OUT
   62 FORMAT(1H ,'UNIT NUMBER FOR OUTPUT DATA=  ',I2)
      L=IN-OUT
      IF(L.NE.4)GOTO 63
      WRITE(2,64)
```

```
   64 FORMAT(1H ,'INPUT RUN NUMBER?...')
      READ(1,65)XX
   65 FORMAT(10A1)
      WRITE(3,66)XX
      WRITE(12,66)XX
   66 FORMAT(1H ,'RUN NUMBER: ',10A1)
      IF(OUT.EQ.5)GOTO 67
      IF(OUT.EQ.6)GOTO 68
      IF(OUT.EQ.7)GOTO 69
   67 WRITE(12,70)
   70 FORMAT(//'DATA FOR: SHELL INLET')
      GOTO 73
   68 WRITE(12,71)
   71 FORMAT(//'DATA FOR: SHELL OUTLET')
      GOTO 73
   69 WRITE(12,72)
   72 FORMAT(//'DATA FOR: TUBE OUTLET')
   73 WRITE(2,450)
  450 FORMAT(25H     POLAR=1,VECTOR=2 ?...)
      READ(1,451)K1
  451 FORMAT(I1)
      IF(K1.EQ.1) GO TO 703
      WRITE(3,702)
  702 FORMAT(15H     VECTOR DATA)
      GOTO 15
  703 WRITE(3,704)
  704 FORMAT(14H     POLAR DATA)
   80 IF(J2.EQ.0)GOTO 15
      REWIND IN
      WRITE(2,81)
   81 FORMAT(1H ,'DELETE PREVIOUS RESULTS? NO=0; YES=1:..')
      READ(1,251)J1
      WRITE(3,82)J1
   82 FORMAT(1H ,'PREVIOUS RESULTS DELETED: NO=0; YES=1: ',I1)
      IF(J1.EQ.0)GOTO 15
      REWIND OUT
      REWIND 12
   15 READ(IN,452)(F(J),R(J),I(J),J=1,K)
  452 FORMAT(3(1X,F12.5))
      IF(J2.GT.0)GOTO 24
      WRITE(3,599)
  599 FORMAT(25H      ......INPUT DATA......)
      WRITE(3,452)(F(J),R(J),I(J),J=1,K)
   24 WRITE(2,4)
    4 FORMAT(29H     NO.OF FFIT ITERATIONS?...)
C IF IT1 IS +VE,FFIT ITERATES UNTIL CONVERGENCE LIMIT MET:
C IF IT1 IS -VE,FFIT ITERATES FOR MOD(IT1) ITERATIONS
      READ(1,500)IT1
  500 FORMAT(I6)
      WRITE(3,705)IT1
  705 FORMAT(27H     NO.OF FFIT ITERATIONS =,I6)
      IF(IT1)34,35,35
   35 WRITE(2,36)
   36 FORMAT(25H     CONVERGENCE LIMIT?...)
      READ(1,501)FIT
  501 FORMAT(1PE12.4)
      WRITE(3,706)FIT
  706 FORMAT(23H     CONVERGENCE LIMIT =,1PE12.4)
   34 WRITE(2,37)
   37 FORMAT(37H     ORDER OF NUMERATOR POLYNOMIAL?...)
      READ(1,502)NP
  502 FORMAT(I2)
```

```
      WRITE(3,707)NP
  707 FORMAT(17H    NUMER.ORDER =,I2)
      WRITE(2,503)
  503 FORMAT(39H    ORDER OF DENOMINATOR POLYNOMIAL?...)
      READ(1,504)MP
  504 FORMAT(I2)
      WRITE(3,708)MP
  708 FORMAT(17H    DENOM.ORDER =,I2)
      KM=NP+MP+1
      IF(KM*(KM+5).GT.400) GO TO 34
      IF(K1.EQ.1) GO TO 850
      GO TO 851
  850 WRITE(3,852)
  852 FORMAT(14H    ENTER RIMP)
      IF(K1.EQ.1) CALL RIMP(R,I,K,-1)
      WRITE(3,853)
  853 FORMAT(13H    EXIT RIMP)
  851 IT=IT1
      WRITE(3,709)
  709 FORMAT(14H    ENTER FFIT)
      CALL FFIT(F,R,I,K,SPN,SPD,400,FIT,IT,NP,MP)
      WRITE(3,710)
  710 FORMAT(13H    EXIT FFIT)
      IF(IT.EQ.-1) WRITE(2,507)FIT
      IF(IT.EQ.-1) WRITE(3,507)FIT
  507 FORMAT(20H    CRASH EXIT: FIT=,1PE12.4)
      J2=J2+1
      IF(IT.EQ.-1) GOTO 80
      WRITE(2,5)FIT
      WRITE(3,5)FIT
      WRITE(12,74)
      WRITE(12,5)FIT
    5 FORMAT(37H    SUM OF SQUARES OF ERROR OF FIT = ,1PE12.4)
      ND=IFIX(SPD(1))
      NN=IFIX(SPN(1))
      DO 11 J=1,ND
   11 SPD(J)=SPD(J+1)
      DO 12 J=1,NN
   12 SPN(J)=SPN(J+1)
      G1=SPN(NN)/SPD(ND)
   42 WRITE(3,6)
      WRITE(2,6)
      WRITE(12,74)
      WRITE(12,6)
    6 FORMAT(36H.....0-N...........DENOM COEFFS.....)
      DO 7 J=1,ND
      JJ=J-1
      WRITE(3,8)JJ,SPD(J)
      WRITE(12,8)JJ,SPD(J)
    7 WRITE(2,8)JJ,SPD(J)
    8 FORMAT(5X,I3,18X,1PE12.4)
      WRITE(3,9)
      WRITE(2,9)
      WRITE(12,74)
      WRITE(12,9)
    9 FORMAT(36H....0-N............NUMER COEFFS.....)
      DO 10 J=1,NN
      JJ=J-1
      WRITE(3,8)JJ,SPN(J)
      WRITE(12,8)JJ,SPN(J)
   10 WRITE(2,8)JJ,SPN(J)
      IF(ND.EQ.1) GO TO 27
```

```
      ITER=1000
      CRIT=.0001
   30 J=ND-1
      WRITE(3,711)
  711 FORMAT(15H    ENTER NEWER)
      CALL NEWER(SPD,J,R,ITER,CRIT)
      WRITE(3,712)
  712 FORMAT(14H    EXIT NEWER)
      IF(J)28,29,28
   28 WRITE(2,3)ITER,CRIT
      WRITE(3,3)ITER,CRIT
      WRITE(2,87)
   87 FORMAT(1H ,'NEW ITER=...')
      READ(1,505)ITER
  505 FORMAT(I6)
      WRITE(2,88)
   88 FORMAT(1H ,'NEW CRIT=...')
      READ(1,506)CRIT
  506 FORMAT(F7.5)
      WRITE(3,89)ITER,CRIT
   89 FORMAT(1H ,'NEW ITER= ',I6,'  NEW CRIT= ',F7.5)
      GO TO 30
   29 J=ND-J-1
      JD=J
      J=J+J
      WRITE(3,25)
      WRITE(12,74)
   74 FORMAT(1H )
      WRITE(12,25)
   25 FORMAT(36H......REAL POLE.......IMAG POLE.....)
      WRITE(3,13)(R(I1),R(I1+1),I1=1,J,2)
      WRITE(12,13)(R(I1),R(I1+1),I1=1,J,2)
      WRITE(OUT,40)JD,(R(I1),R(I1+1),I1=1,J,2)
   40 FORMAT(I3/12(1PE12.4))
      CALL GAIN2(R,JD,G2)
      GOTO 50
   27 G2=1.0
      JD=0
      R(1)=0.0
      R(2)=0.0
      WRITE(OUT,40)JD,R(1),R(2)
   50 IF(NN.EQ.1) GO TO 130
      ITER=1000
      CRIT=.0001
   31 J=NN-1
      WRITE(3,713)
  713 FORMAT(15H    ENTER NEWER)
      CALL NEWER(SPN,J,R,ITER,CRIT)
      WRITE(3,714)
  714 FORMAT(14H    EXIT NEWER)
      IF(J)32,33,32
   32 WRITE(2,3)ITER,CRIT
      WRITE(3,3)ITER,CRIT
      WRITE(2,87)
      READ(1,505)ITER
      WRITE(2,88)
      READ(1,506)CRIT
      WRITE(3,89)ITER,CRIT
      GO TO 31
   33 J=NN-J-1
      JN=J
      J=J+J
```

```
  3 FORMAT(32H      ROOTS NOT FOUND WITH ITER= ,I6,7H CRIT= ,
   1E8.1,24H     ENTER NEW ITER,CRIT )
    WRITE(3,26)
    WRITE(12,74)
    WRITE(12,26)
 26 FORMAT(34H....REAL ZERO.......IMAG ZERO.....)
    WRITE(3,13)(R(I1),R(I1+1),I1=1,J,2)
    WRITE(12,13)(R(I1),R(I1+1),I1=1,J,2)
 13 FORMAT(4X,1PE12.4,5X,1PE12.4)
    WRITE(OUT,40)JN,(R(I1),R(I1+1),I1=1,J,2)
    CALL GAIN2(R,JN,G3)
    GOTO 51
130 JN=0
    G3=1.0
    R(1)=0.0
    R(2)=0.0
    WRITE(OUT,40)JN,R(1),R(2)
 51 G=G1*G3/G2
    WRITE(3,43)G
    WRITE(3,600)G1
    WRITE(12,600)G1
    WRITE(12,43)G
    WRITE(OUT,44)G1
600 FORMAT(/'GAIN=  ',F12.5)
 43 FORMAT(/'STEADY STATE GAIN=   ',F12.5)
 44 FORMAT(F12.5)
    WRITE(2,23)
    WRITE(3,23)
 23 FORMAT(1H ,'FIT TO SAME DATA? NO=0: YES=1:...')
    READ(1,508)J
508 FORMAT(I2)
    WRITE(3,86)J
 86 FORMAT(1H ,'FIT TO SAME DATA? NO=0: YES=1:...',I2)
    J2=J2+1
    IF(J-1)99,80,80
 99 STOP
    END
```

```
C    SUBROUTINE GAIN2 IS A STEP IN THE CALCULATION OF THE
C    STEADY STATE GAIN.PERFORMS THE CALCULATION-
C    PRODUCT OF POLES OR ZEROS.
C
      SUBROUTINE GAIN2(B,N1,G)
      DIMENSION A(12),G1(6),B(50)
      WRITE(3,5)
    5 FORMAT('     ENTER GAIN2')
      W=0.001
      IF(N1.EQ.1)GOTO 40
      N2=2*N1
      DO 1 I=1,N2
    1 A(I)=-B(I)
      DO 7 I=1,N2,2
    7 IF(A(I).EQ.0.0000)A(I)=W
      M=(N1/2)*2
      IF(M.EQ.N1)GOTO 10
      A(N2+1)=1.0
      A(N2+2)=0.0
      N2=N2+2
   10 J=0
      N3=N2+1
      DO 20 I=1,N3,4
      J=J+1
   20 G1(J)=A(I)*A(I+2)-A(I+1)*A(I+3)
      G=1.0
      J1=N2/4
      DO 30 J=1,J1
   30 G=G*G1(J)
      GOTO 50
   40 G=-B(1)
   50 WRITE(3,6)
    6 FORMAT('     EXIT GAIN2')
      RETURN
      END
```

A2.2.4. PROGRAMME 'ESTOCS'

```
C     PROGRAM:ESTOCS
C     MASTER PROGRAM FOR OBTAINING C(S) FROM E(S):C(S)=R(S)-E(S)
C     THE INPUT TO ESTOCS IS THE OUTPUT FROM PROGRAM FITFRQ
C
C     ESTOCS CALLS THE FOLLOWING ROUTINES:
C     POLMUL.....POLYNOMIAL COEFFICIENTS FROM ITS ZEROS ROUTINE
C     POLADD.....ADDITION OF TWO POLYNOMIALS ROUTINE
C     ROOTS......(MODIFICATION OF NEWER - SEE FITFRQ)
C     POLVAL.....(ROUTINE CALLED BY ROOTS)
C     ITEX.......NEW ITERATION & CONVERGENCE LIMIT ROUTINE
C     GAIN.......ROUTINE USED IN THE CALCULATION OF THE STEADY
C                STATE GAIN
C     OUT........DATA OUTPUT ROUTINE
C
C     LANGUAGE...FORTRAN IV
C
C     UNIT NUMBERS ASSIGNED TO:
C     1...USER TELETYPE FOR READING OF INPUT PARAMETERS REQUESTED BY
C         PROGRAM
C     2...VDU DISPLAY
C     3...UNASSIGNED
C     4...FILE CONTAINING SUBROUTINE ENTER&EXIT STATEMENTS-AID TO
C         FAULT FINDING
C     5...SHELL INLET DATA FILE
C     6...SHELL OUTLET DATA FILE
C     7...TUBE OUTLET DATA FILE
C     8...OUTPUT DATA FILE
C     UNITS 4-8 ARE FILES ON THE 'SCRATCH DISC'
C
      DIMENSION ESHIN1(12),ESHID1(12),CSHIN(14),CSHID(14),POLSI1(16)
      DIMENSION POLSI2(16),POLSI3(16),XX(10)
      INTEGER ESHIZ1,ESHIZ2,ESHIP1,CSHIZ,CSHIP
      REAL KO,K1,K2
      WRITE(4,5)
    5 FORMAT('     ENTER ESTOCS')
      K=0
      WRITE(2,47)
      WRITE(4,47)
   47 FORMAT(1H ,'INPUT RUN NUMBER?...'/)
      READ(1,48)XX
   48 FORMAT(10A1)
      WRITE(4,49)XX
      WRITE(8,49)XX
   49 FORMAT(1H ,'RUN NUMBER: ',10A1)
  200 DO 1 I=1,12
      ESHIN1(I)=0.0
      ESHID1(I)=0.0
      CSHIN(I)=0.0
    1 CSHID(I)=0.0
      K=K+1
      GOTO(201,202,203)K
  201 WRITE(2,210)
  210 FORMAT(1H ,'INITIAL SHELL INLET TEMP: DEG C ?...')
      READ(1,205)T1
      WRITE(2,211)
  211 FORMAT(1H ,'FINAL SHELL INLET TEMP: DEG C ?...')
      READ(1,205)T2
      KO=T2-T1
      WRITE(8,215)KO
  215 FORMAT(1H ,'MAGNITUDE OF STEP INPUT: ',F12.5,' DEG C')
      GOTO 204
  202 WRITE(2,212)
```

```
212 FORMAT(1H ,'INITIAL SHELL OUTLET TEMP; DEG C ?...')
    READ(1,205)T1
    WRITE(2,213)
213 FORMAT(1H ,'FINAL SHELL OUTLET TEMP; DEG C ?...')
    READ(1,205)T2
    GOTO 204
203 WRITE(2,214)
214 FORMAT(1H ,'INITIAL TUBE OUTLET TEMP; DEG C ?...')
    READ(1,205)T1
    WRITE(2,216)
216 FORMAT(1H ,'FINAL SHELL OUTLET TEMP; DEG C ?...')
    READ(1,205)T2
205 FORMAT(F5.2)
204 WRITE(4,206)T1,T2
206 FORMAT('T1= ',F5.2,' T2= ',F5.2)
    KO=T2-T1
    IN=4+K
    READ(IN,45)ESHIP1,(ESHID1(I),I=1,12),ESHIZ1,
   1(ESHIN1(I),I=1,12),K1
    WRITE(4,50)ESHIP1,(ESHID1(I),I=1,12),ESHIZ1,
   1(ESHIN1(I),I=1,12),K1
 45 FORMAT(2(I3/12(1PE12.4)/),0PF12.5)
 50 FORMAT(1H ,2(2X,I3/6(5X,1PE12.4,5X,1PE12.4/)),2X,0PF12.5)
    DO 55 I=1,12
    ESHID1(I)=-ESHID1(I)
 55 ESHIN1(I)=-ESHIN1(I)
    DO 2 I=1,16
    POLSI1(I)=0.0
    POLSI2(I)=0.0
  2 POLSI3(I)=0.0
    IF(ESHIZ1.GT.6.OR.ESHIP1.GT.6)GOTO 95
    IF(ESHIP1.EQ.0)GOTO 7
    M1=2*(ESHIP1+1)
    CALL POLMUL(ESHID1,ESHIP1,POLSI1,M1)
    DO 10 I=1,M1
 10 POLSI1(I)=KO*POLSI1(I)
    GOTO 12
  7 POLSI1(1)=KO
    POLSI1(2)=0.0
 12 WRITE(4,100)
100 FORMAT(///'KO*POLSI1(I)=')
    DO 101 I=1,16,2
101 WRITE(4,102)POLSI1(I),POLSI1(I+1)
102 FORMAT(1H ,2(5X,F10.5))
    IF(ESHIZ1.EQ.0)GOTO 17
    M2=2*(ESHIZ1+1)
    CALL POLMUL(ESHIN1,ESHIZ1,POLSI2,M2)
    GOTO 18
 17 POLSI2(1)=1.0
    POLSI2(2)=0.0
    M2=2
 18 ESHIZ2=ESHIZ1+1
    POLSI2(M2+1)=0.0
    POLSI2(M2+2)=0.0
    M3=M2+2
    DO 20 I=1,M3
 20 POLSI2(I)=-K1*POLSI2(I)
 19 WRITE(4,103)
103 FORMAT(///'-K1*POLSI2(I)=')
    DO 104 I=1,16,2
104 WRITE(4,102)POLSI2(I),POLSI2(I+1)
    N3=ESHIP1
```

```
      IF(ESHIP1.LT.ESHIZ2)N3=ESHIZ2
      N4=2*N3+2
      CSHIZ=N3
      CALL POLADD(POLSI1,ESHIP1,POLSI2,ESHIZ2,POLSI3,N3)
      G1=POLSI3(N4-1)
      DO 25 I=1,N4
   25 POLSI3(I)=POLSI3(I)/G1
      WRITE(4,105)G1
  105 FORMAT(///'G1=   ',F12.5)
      WRITE(4,106)
  106 FORMAT(///'POLSI3(I)/G1=')
      DO 107 I=1,16,2
  107 WRITE(4,102)POLSI3(I),POLSI3(I+1)
      ITER=1000
      CRIT=0.0001
   23 CALL ROOTS(POLSI3,N3,CSHIN,ITER,CRIT)
      IF(N3)21,22,21
   21 CALL ITEX(ITER,CRIT)
      GOTO 23
   22 CALL GAIN(CSHIN,CSHIZ,G2,0.001)
      J=0
      WRITE(4,108)G2
  108 FORMAT(///'G2=   ',F12.5)
      WRITE(4,109)
  109 FORMAT(///'CSHIN=')
      DO 110 I=1,14,2
  110 WRITE(4,102)CSHIN(I),CSHIN(I+1)
      IF(ESHIP1.EQ.0)GOTO 35
      J=2*ESHIP1
      DO 30 I=1,J
   30 CSHID(I)=-ESHID1(I)
      CALL GAIN(CSHID,ESHIP1,G3,0.001)
   35 IF(ESHIP1.EQ.0)G3=1.0
      K2=(G1*G2/G3)
      CSHID(J+1)=0.0
      CSHID(J+2)=0.0
      CSHIP=ESHIP1+1
      WRITE(4,111)
  111 FORMAT(///'CSHID=')
      DO 112 I=1,14,2
  112 WRITE(4,102)CSHID(I),CSHID(I+1)
      L1=2*CSHIZ
      L2=2*CSHIP
      CALL OUT(CSHIN,L1,CSHID,L2,G1,K2,K,T1,T2)
      IF(K.EQ.3)GOTO 97
      GOTO 200
   95 WRITE(2,96)
      WRITE(4,96)
   96 FORMAT(' NUMBER OF POLES OR ZEROS IN INPUT EXCEEDS THAT SPECIFIED
     1 IN PROGRAM')
   97 WRITE(4,6)
    6 FORMAT('     EXIT ESTOCS')
      STOP
      END
```

```
C     SUBROUTINE POLMUL CONSTRUCTS THE COEFFICIENTS OF A
C     POLYNOMIAL FROM ITS REAL &/OR COMPLEX ZEROS
C
      SUBROUTINE POLMUL(G,N,H,N3)
      DIMENSION A(14),B(14),C(2),D(2),E(14),F(14),G(24),H(26)
      WRITE(4,5)
    5 FORMAT(16H    ENTER POLMUL)
      N2=2*N
      DO 1 I=1,14
      A(I)=0.0
    1 B(I)=0.0
      DO 2 I=1,26
    2 H(I)=0.0
      A(1)=1.0
      C(1)=1.0
      D(1)=0.0
      A(2)=G(1)
      B(2)=G(2)
      NDA=0
      DO 40 K=4,N2,2
      C(2)=G(K-1)
      D(2)=G(K)
      NDA=NDA+1
      NDE=NDA+1
      NDAP1=NDE
      NDEP1=NDE+1
      DO 20 I=1,NDEP1
      E(I)=0.0
   20 F(I)=0.0
      DO 10 I=1,NDAP1
      DO 15 J=1,2
      IPJ=I+J-1
      E(IPJ)=E(IPJ)+A(I)*C(J)-B(I)*D(J)
      F(IPJ)=F(IPJ)+B(I)*C(J)+A(I)*D(J)
   15 CONTINUE
   10 CONTINUE
      DO 25 I=1,NDEP1
      A(I)=0.0
   25 B(I)=0.0
      DO 30 I=1,NDEP1
      A(I)=E(I)
   30 B(I)=F(I)
   40 CONTINUE
      N3=2*NDEP1
      J=0
      DO 50 I=1,NDEP1
      J=J+2
      H(J-1)=A(I)
   50 H(J)=B(I)
      WRITE(4,6)
    6 FORMAT(15H    EXIT POLMUL)
      RETURN
      END
```

```
C     SUBROUTINE POLADD PERFORMS ADDITION OF TWO POLYNOMIALS WITH
C     REAL & COMLEX TERMS
C
      SUBROUTINE POLADD(X,NDA,Y,NDC,H,NDE)
      DIMENSION A(14),B(14),C(14),D(14),E(14),F(14),X(26),Y(26),
     1H(26)
      WRITE(4,5)
    5 FORMAT(16H     ENTER POLADD)
      DO 3 I=1,14
      A(I)=0.0
      B(I)=0.0
      C(I)=0.0
      D(I)=0.0
      E(I)=0.0
    3 F(I)=0.0
      DO 4 I=1,26
    4 H(I)=0.0
      N1=NDA+1
      N2=NDC+1
      N3=NDE+1
      J1=2*N1
      J2=2*N2
      J3=2*N3
      J=J1+2
      DO 1 I=1,N1
      J=J-2
      A(I)=X(J-1)
    1 B(I)=X(J)
      J=J2+2
      DO 2 I=1,N2
      J=J-2
      C(I)=Y(J-1)
    2 D(I)=Y(J)
      IF(NDA.EQ.NDC)GOTO 60
      IF(NDA.LT.NDC)GOTO 70
      IF(NDA.GT.NDC)GOTO 80
   60 DO 10 I=1,N1
      E(I)=A(I)+C(I)
   10 F(I)=B(I)+D(I)
      GOTO 90
   70 DO 20 I=1,N1
      E(I)=A(I)+C(I)
   20 F(I)=B(I)+D(I)
      M1=NDA+2
      DO 30 I=M1,N2
      E(I)=C(I)
   30 F(I)=D(I)
      GOTO 90
   80 DO 40 I=1,N2
      E(I)=A(I)+C(I)
   40 F(I)=B(I)+D(I)
      M2=NDC+2
      DO 50 I=M2,N1
      E(I)=A(I)
   50 F(I)=B(I)
   90 J=0
      DO 95 I=1,N3
      J=J+2
      H(J)=F(I)
   95 H(J-1)=E(I)
      WRITE(4,6)
    6 FORMAT(15H     EXIT POLADD)
      RETURN
      END
```

```
C SUBROUTINE TO FIND ROOTS OF POLYNOMIAL:
C AND HENCE FIND THE POLES AND ZEROS OF THE NUMERATOR
C AND DENOMINATOR POLYNOMIALS
      SUBROUTINE ROOTS(AA,N,B,ITN,ERR)
      LOGICAL FLAG
      REAL AA(26),A(13),B(24),NUMER,NUMR,NUMI
      EQUIVALENCE (DENOM,DENR),(NUMER,NUMR,BTMP)
      WRITE(4,100)
  100 FORMAT('     ENTER ROOTS')
      J=0
      DO 90 I=1,26,2
      J=J+1
   90 A(J)=AA(I)
      N1=N+N
      IF(N.LT.2) GO TO 21
      INC=ALOG10(ABS(A(1)/A(N+1)))/(N-1)
      N2=N
      DO 21 J=0,N2
      A(J+1)=A(J+1)*10.0**(INC*(J+1)-1)
   21 CONTINUE
      J=1
    3 IF(A(1).NE.0.0) GO TO 1
      DO 2 I=1,N
    2 A(I)=A(I+1)
      B(J)=0.0
      J=J+1
      N=N-1
      GO TO 3
    1 I=0
      IF (N-2)10,8,18
   18 B(J)=0.0
      B(J+1)=0.0
    5 I=I+1
      FLAG=.FALSE.
      WRITE(4,200)
  200 FORMAT(16H     ENTER POLVAL)
      CALL POLVAL(A,N,B(J),B(J+1),NUMER,NUMI,FLAG)
      WRITE(4,201)
  201 FORMAT(15H     EXIT POLVAL)
      IF (ABS(NUMER).LE.ERR) GO TO 4
      FLAG=.TRUE.
      WRITE(4,200)
      CALL POLVAL(A,N,B(J),B(J+1),DENOM,DENI,FLAG)
      WRITE(4,201)
      B(J)=B(J)-NUMER/DENOM
      IF (I.LE.ITN) GO TO 5
      GO TO 12
    4 DO 6 I=1,N
    6 A(N+1-I)=A(N+1-I)+A(N+2-I)*B(J)
      DO 7 I=1,N
    7 A(I)=A(I+1)
      N=N-1
      J=J+2
      GO TO 1
   10 B(J)=-A(1)/A(2)
      N=N-1
      J=J+1
      B(J)=0.0
      GO TO 20
   12 I=0
      B(J)=1.0
      B(J+1)=1.0
```

```
11 I=I+1
   FLAG=.FALSE.
   WRITE(4,200)
   CALL POLVAL(A,N,B(J),B(J+1),NUMR,NUMI,FLAG)
   WRITE(4,201)
   IF (SQRT(NUMR*NUMR+NUMI*NUMI).LE.ERR) GO TO 9
   FLAG=.TRUE.
   WRITE(4,200)
   CALL POLVAL(A,N,B(J),B(J+1),DENR,DENI,FLAG)
   WRITE(4,201)
   ATMP=DENR*DENR+DENI*DENI
   B(J)=B(J)-(NUMR*DENR+NUMI*DENI)/ATMP
   B(J+1)=B(J+1)+(NUMR*DENI-NUMI*DENR)/ATMP
   IF (I.EQ.ITN) GO TO 20
   GO TO 11
 9 B(J+2)=B(J)
   B(J+3)=-B(J+1)
   ATMP=B(J)+B(J)
   BTMP=B(J)*B(J)+B(J+1)*B(J+1)
   J=J+4
   NM1=N-1
   DO 13 I=1,NM1
   A(N-I+1)=A(N-I+1)+ATMP*A(N-I+2)
13 A(N-I)=A(N-I)-BTMP*A(N-I+2)
   DO 14 I=1,NM1
14 A(I)=A(I+2)
   N=N-2
   IF (N.GE.4) GO TO 12
   IF (N.EQ.3) GO TO 1
 8 B(J)=-0.5*A(2)/A(3)
   B(J+2)=B(J)
   ATMP=A(2)*A(2)-4.0*A(3)*A(1)
   B(J+1)=0.0
   B(J+3)=0.0
   IF (ATMP)15,16,17
15 ATMP=SQRT(-ATMP)*0.5/A(3)
   B(J+1)=ATMP
   B(J+3)=-ATMP
16 N=N-2
   GO TO 20
17 ATMP=SQRT(ATMP)*0.5/A(3)
   B(J)=B(J)+ATMP
   B(J+2)=B(J+2)-ATMP
   GO TO 16
20 DO 19 J=1,N1
19 B(J)=B(J)*10.0**INC
   WRITE(4,210)
210 FORMAT(23H     B(J)=B(J)*10.0**INC)
   WRITE(4,211) (B(J),J=1,N1)
211 FORMAT(5X,E12.4)
   DO 22 J=0,N2
   A(J+1)=(A(J+1)*10.0)**(-(INC*(J+1)-1))
22 CONTINUE
   WRITE(4,212)
212 FORMAT(41H     A(J+1)=(A(J+1)*10.0)**(-(INC(J+1)-1)))
   WRITE(4,888) (A(J+1),J=0,N2)
888 FORMAT(5X,E12.4)
   WRITE(4,101)
101 FORMAT('     EXIT ROOTS')
   RETURN
   END
```

```
C SUBROUTINE CALLED BY 'ROOTS'
      SUBROUTINE POLVAL(A,N,SR,SI,VALR,VALI,FLAG)
      INTEGER ORDER
      LOGICAL FLAG
      REAL A(400),SR,SI,VALR,VALI,ATMPR
      ORDER=N
      IF (FLAG) ORDER=N-1
      VALR=0.0
      VALI=0.0
      DO 1 I=0,ORDER
      ATMPR=A(N-I+1)
      IF (FLAG) ATMPR=ATMPR*(N-I)
      ATMPR=VALR*SR-VALI*SI+ATMPR
      VALI=VALR*SI+VALI*SR
    1 VALR=ATMPR
      WRITE(4,200)VALR,VALI
  200 FORMAT(9H    VALR=,E12.4,9H    VALI=,E12.4)
      RETURN
      END
```

```
C   SUBROUTINE ITEX IS USED IN CONJUNCTION WITH SUBROUTINE ROOTS
C   TO ENTER NEW NUMBER OF ITERATIONS & NEW CONVERGENCE LIMIT
C
      SUBROUTINE ITEX(ITER,CRIT)
      WRITE(4,5)
    5 FORMAT(15H     ENTER ITEX)
      WRITE(2,10)ITER,CRIT
      WRITE(4,10)ITER,CRIT
   10 FORMAT(32H     ROOTS NOT FOUND WITH ITER= ,I6,9H   CRIT= ,E8.1)
      WRITE(1,15)
   15 FORMAT(1H ,'INPUT NEW NUMBER OF ITERATIONS?...')
      READ(1,20)ITER
      WRITE(4,25)ITER
   20 FORMAT(I6)
   25 FORMAT(1H ,'NEW ITER=  ',I6)
      WRITE(1,26)
   26 FORMAT(1H ,'INPUT NEW CONVERGENCE LIMIT?...')
      READ(1,30)CRIT
      WRITE(4,35)CRIT
   30 FORMAT(F7.5)
   35 FORMAT(1H ,'NEW CRIT=  ',F7.5)
      WRITE(4,36)
   36 FORMAT(14H     EXIT ITEX)
      RETURN
      END
```

```
C     SUBROUTINE GAIN IS A STEP IN THE CALCULATION OF THE
C     STEADY STATE GAIN.PERFORMS THE CALCULATION-
C     PRODUCT OF POLES OR ZEROS.
C
      SUBROUTINE GAIN(B,N1,G,W)
      DIMENSION A(24),G1(6),B(24)
      WRITE(4,5)
    5 FORMAT('     ENTER GAIN')
      IF(N1.EQ.1)GOTO 40
      N2=2*N1
      DO 1 I=1,N2
    1 A(I)=-B(I)
      DO 7 I=1,N2,2
    7 IF(A(I).EQ.0.0000)A(I)=W
      M=(N1/2)*2
      IF(M.EQ.N1)GOTO 10
      A(N2+1)=1.0
      A(N2+2)=0.0
      N2=N2+2
   10 J=0
      N3=N2+1
      DO 20 I=1,N3,4
      J=J+1
   20 G1(J)=A(I)*A(I+2)-A(I+1)*A(I+3)
      G=1.0
      J1=N2/4
      DO 30 J=1,J1
   30 G=G*G1(J)
      GOTO 50
   40 G=-B(1)
   50 WRITE(4,6)
    6 FORMAT('     EXIT GAIN')
      RETURN
      END
```

```
C     SUBROUTINE OUT CONTROLS THE OUTPUT OF DATA
C
      SUBROUTINE OUT(A,L1,B,L2,GO,G,K,T1,T2)
      DIMENSION A(24),B(24)
      WRITE(4,5)
    5 FORMAT(13H     ENTER OUT)
      WRITE(4,10)
      WRITE(8,10)
   10 FORMAT(///'RESPONSE DATA FOR:')
      GOTO(11,12,13)K
   11 WRITE(4,15)
      WRITE(8,15)
   15 FORMAT('SHELL INLET')
      GOTO 14
   12 WRITE(4,16)
      WRITE(8,16)
   16 FORMAT('SHELL OUTLET')
      GOTO 14
   13 WRITE(4,17)
      WRITE(8,17)
   17 FORMAT('TUBE OUTLET')
   14 WRITE(8,50)T1
   50 FORMAT('INITIAL TEMP= ',F5.2,' DEG C')
      WRITE(8,51)T2
   51 FORMAT('FINAL TEMP= ',F5.2,' DEG C')
      WRITE(8,19)GO
   19 FORMAT(//'GAIN= ',F12.5)
      WRITE(8,20)
   20 FORMAT(//'       REAL ZERO          IMAG ZERO')
      WRITE(8,30)(A(I),A(I+1),I=1,L1,2)
   30 FORMAT(4X,1PE12.4,5X,1PE12.4)
      WRITE(8,40)
   40 FORMAT(//'       REAL POLE          IMAG POLE')
      WRITE(8,30)(B(I),B(I+1),I=1,L2,2)
      WRITE(8,18)G
   18 FORMAT(//'STEADY STATE GAIN= ',F12.5)
      WRITE(4,6)
    6 FORMAT(12H     EXIT OUT)
      RETURN
      END
```

APPENDIX A3

STEADY STATE HEAT TRANSFER TESTS

A3.1. UNCERTAINTY ANALYSIS

The uncertainty in the value of a measurement or the result of a calculation involving several variables is usually sub-divided into the random uncertainty and the systematic uncertainty. The random uncertainty can only be derived by statistical methods from a number of repeated readings. Because sufficient repeat data is not available for the steady state tests the random uncertainty component can not be estimated. Two methods which are used to estimate the total systematic uncertainty are based on the assumption that the variables involved in the determination of the required parameter are independent. Thus, if the value of a particular parameter y is dependant on a number of independent variables x_1, x_2..... x_n, the total systematic uncertainty may be expressed as:

$$\delta y = \sum_{i=1}^{n} \left| \frac{\partial y}{\partial x_i} \right| \delta x_i = \sum_{i=1}^{n} \delta y_i \qquad \text{(A3.1)}$$

where δy_i = the component of the systematic uncertainty associated with y due to the systematic uncertainty δx_i in the variable x_i.

The second method combines the component systematic uncertainties, y_i, in quadrature, i.e.,

$$(\delta y)^2 = \sum_{i=1}^{n} \left(\frac{\partial y}{\partial x_i} \cdot \delta x_i \right)^2 \qquad \text{(A3.2)}$$

The first method, equation A3.1, is likely to over-estimate the total systematic uncertainty because it is unlikely that the maximum systematic uncertainties of the components will occur together. However, this method can be considered as giving a maximum limit to the uncertainty. The second method, equation A3.2., tends to under-estimate the size of the overall systematic uncertainty particularly when one of the components is considerably larger than the others.

A very informative guide to the subject of uncertainties is reference 9.

For the steady state heat transfer tests the overall heat transfer coefficient is calculated from the equation:

$$U = \frac{\dot{m}_h c_h(\theta_{h1} - \theta_{h2}) \text{ OR } \dot{m}_c c_c(\theta_{c2} - \theta_{c1})}{A_o \left[\dfrac{(\theta_{h1} - \theta_{c2}) - (\theta_{h2} - \theta_{c1})}{\ln\left\{\dfrac{\theta_{h1} - \theta_{c2}}{\theta_{h2} - \theta_{c1}}\right\}}\right]} \qquad (A3.3)$$

Because of the inter-relationship between the variables on the right-hand side of this equation, the variables are not independent of each other and, therefore, the methods outlined above to estimate the systematic uncertainty cannot be applied.

In order to allocate limits to the systematic uncertainty, (hereafter called the uncertainty), the approach given below is adopted.

Combining equations I.6 - I.8 with $F_c = 1$:

$$U = \frac{\dot{m}_c c_c \Delta\theta_c}{A_o \Delta\theta_{LMTD}} = \frac{\dot{m}_h c_h \Delta\theta_h}{A_o \Delta\theta_{LMTD}} \qquad (A3.4)$$

Introducing a small change δ into each of the parameters on the right-hand side of equation A3.4 will produce a small change δU in U:

$$\therefore\ (U + \delta U) = \frac{(\dot{m} + \delta\dot{m})\ (c + \delta c)\ (\Delta\theta + \delta\Delta\theta)}{(A_o + \delta A_o)\ (\Delta\theta_{LMTD} + \delta\Delta\theta_{LMTD})} \qquad (A3.5)$$

The percentage uncertainty in U may be given as:

$$\begin{aligned} \delta U(\%) &= \left[\frac{(U + \delta U) - U}{U}\right] \times 100\ \% \\ &= \left[\frac{(U + \delta U)}{U} - 1\right] \times 100\ \% \end{aligned} \qquad (A3.6)$$

Substituting equation A3.5 into A3.6 and dividing by U gives:

$$U(\%) = \left[\frac{(1 + x_{\dot{m}})\,(1 + x_c)\,(1 + x_{\Delta\theta})}{(1 + x_{A_o})\,(1 + x_{\Delta\theta_{LMTD}})} - 1 \right] \times 100 \qquad (A3.7)$$

where $x_{\dot{m}} = \frac{\delta \dot{m}}{\dot{m}}$, $x_c = \frac{\delta c}{c}$ etc.,

The maximum positive uncertainty occurs when the component uncertainties in the numerator of equation A3.7 are positive and those in the denominator negative. Conversely the maximum negative uncertainty is when the numerator component uncertainties are negative and those in the denominator positive. When actual values are substituted into equation A3.7 for the above two cases a slightly asymmetrical uncertainty results. Because the maximum positive uncertainty is always greater than the maximum negative uncertainty, the uncertainty in U is taken as plus and minus the maximum positive uncertainty.

The problem when attempting to apply equation A3.7 is in assigning realistic values to the component uncertainties. The estimated value of the uncertainty in each component and the assumptions made in its determination are given below.

MAXIMUM COMPONENT UNCERTAINTIES

(i) In the flow rates:

(a) The water flow rate calibration curve was generated from B.S. 1042 Part 1: 1964 and verified by deadweight calibration. Agreement was better than $\pm$ 1%. Therefore, assume a tolerance on the water flow rate of $\pm$ 1%.

(b) The oil flow rate calibration was determined by deadweight measurement and a volumetric flow rate calculation. The scatter band on the calibration curve was $< \pm 0.5\%$. Therefore assume a tolerance on the oil flow calibration of $\pm 0.5\%$.

(ii) In the specific heats:

(a) For the specific heat of water no tolerance figure was quoted in the Steam Tables. The scatter band on the curve fit was $< \pm 0.06\%$. Assume tolerance of $\pm 0.1\%$.

(b) For oil, Shell give the tolerance on the specific heat as $\pm 2\%$.

(iii) In temperature measurement:

Bearing in mind that the thermometers are class A thermometers and can be read to ± 0.05K and appropriate precautions had been taken to ensure confidence in the temperature measurement, i.e., good mixing, the tolerance on each temperature measurement is assumed to be ± 0.05K. (The uncertainty in the temperature difference resulting from the subtraction of two temperatures is $\pm$ 0.1K).

(iv) In the logarithmic mean temperature difference:

It is assumed that the equation for the LMTD in pure counter flow is valid. Using the tolerance of ± 0.05K on each temperature measurement and the actual temperatures recorded in both the water/water and the oil/water tests the maximum variation in the LMTD was $< \pm 1\%$. Therefore, assume an uncertainty in the LMTD of $\pm 1\%$.

(v) In the heat transfer surface area:

The surface area is assumed to be the total tube outside surface area between the tube plates minus the non-effective areas of the tubes at the tube/ baffle hold interfaces. Manufacturing tolerances lead to a maximum tolerance on the surface area of $\pm$ 2%.

ESTIMATED MAXIMUM UNCERTAINTY IN THE OVERALL HEAT TRANSFER COEFFICIENT

The evaluation of the overall heat transfer coefficient and its associated estimated maximum uncertainty is based on either the shell or tubeside heat dissipation rate depending for which calculation the uncertainty in the dissipation rate is the smaller.

For the water/water tests this is the side with the greatest temperature difference. Substituting the component uncertainties into equation A3.7 for the worst case yields the maximum uncertainty in U for the water/water tests as:

$$U(\%) = \pm \left[\frac{(1 + 0.01)(1 + 0.001)(1 + \frac{0.1}{\Delta\theta})}{(1-0.02)(1-0.01)} - 1 \right] \times 100$$

$$= \pm \left[1.0420635 \quad (1 + \frac{0.1}{\Delta\theta}) \quad -1 \right] \times 100 \qquad (A3.8)$$

where $\Delta\theta$ = the temperature difference on the side with the greatest $\Delta\theta$.

For oil/water tests the uncertainty in the oil and water dissipations can be expressed in a modified form as:

$$\text{uncertainty in } Q_{oil} = (1 + 0.005)(1 + 0.02)(1 + \frac{0.1}{\Delta\theta_h}) \qquad (A3.9)$$

$$\text{uncertainty in } Q_{water} = (1 + 0.01)(1 + 0.001)(1 + \frac{0.1}{\Delta\theta_c}) \qquad (A3.10)$$

Dividing A3.9 by A3.10 gives:

$$\frac{Q_{oil}}{Q_{water}} = 1.01394 \left[\frac{1 + \dfrac{0.1}{\Delta\theta_h}}{1 + \dfrac{0.1}{\Delta\theta_c}} \right] \qquad \text{(A3.11)}$$

If this ratio is $\geqslant 1.0$ the evaluation of U is based on the waterside dissipation and if < 1.0 on the oilside dissipation. Rearranging equation A3.11 results in the following criterion to determine for which dissipation the associated uncertainty is the smaller and on which the calculation of U should be based:

$$Y = \frac{\dfrac{1}{\Delta\theta_c}}{0.1394 + \dfrac{1.01394}{\Delta\theta_h}} \left.\begin{matrix} \leqslant 1 \text{ base on } Q_{water} \\ \\ > 1 \text{ base on } Q_{oil} \end{matrix}\right. \qquad \text{(A3.12)}$$

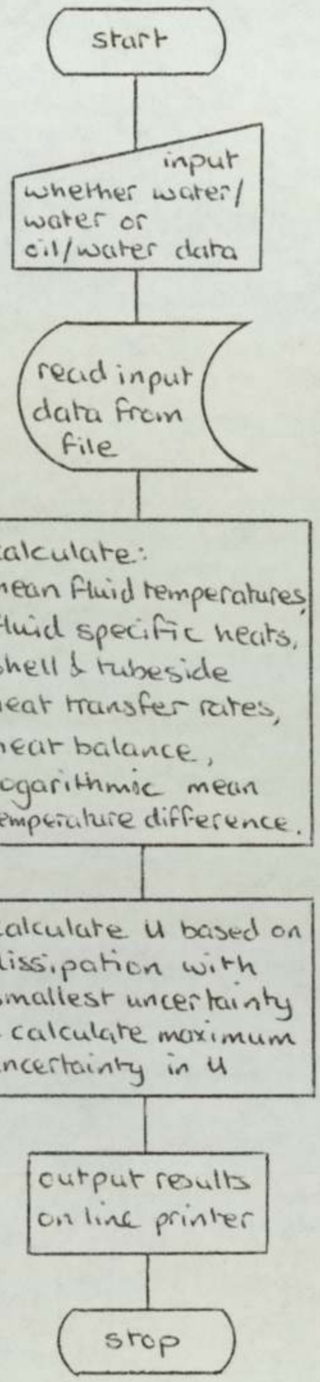
When the uncertainty in U is based on Q_{water}, $\delta U(\%)$ is given by equation A3.8 with $\Delta\theta = \Delta\theta_c$ and when based on Q_{oil} is:

$$U(\%) = \pm \left[\frac{1.005 \times 1.02}{0.98 \times 0.99} \left(1 + \frac{0.1}{\Delta\theta_h}\right) - 1 \right] \times 100$$

$$= \pm \left[1.0565863 \left(1 + \frac{0.1}{\Delta\theta_h}\right) - 1 \right] \text{X } 100 \qquad \text{(A3.13)}$$

Equations A3.8, A3.12 and A3.13 are incorporated in computer programme 'HEAT'.

FIG. A3.1 PROGRAMME 'HEAT' FLOW CHART



EXECUTION: Programme 'HEAT' was run on the Honeywell Mark III Timesharing Computer System.

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```
C PROGRAM HEAT-PERFORMS HEAT TRANSFER ANAYSIS OF STEADY STATE
C TEST RESULTS FOR TEST HEAT EXCHANGER-COUNTER FLOW OPERATION.
C LANGUAGE-F77
      DIMENSION A(50,11)
      STRING B(50)
      REAL MS,MT
      PRINT 10
   10 FORMAT(1X,75HSTEADY STATE HEAT TRANSFER ANALYSIS OF TEST
     &RESULTS FOR TEST HEAT EXCHANGER)
      PRINT,'WATER/WATER DATA INPUT 1;OIL/WATER DATA INPUT 2 '
      INPUT,K
      READ (FILE='H',FMT=*,END=20)((A(I,J),J=1,6),I=1,50)
   20 K1=I-1
      I=0
   30 I=I+1
      TS=(A(I,1)+A(I,2))/2.0
      TT=(A(I,3)+A(I,4))/2.0
      MS=A(I,5)/7938.0
      MT=A(I,6)/7938.0
      A(I,5)=MS
      A(I,6)=MT
      T=TS
      IF(K.LT.1.5)THEN
      CALL PROP (T,C)
      ELSE
      C=1.80481+0.0036275*TS
      END IF
      CS=C
      T=TT
      CALL PROP (T,C)
      CT=C
      T1=A(I,1)-A(I,2)
      T2=A(I,4)-A(I,3)
      QS=MS*CS*T1
      QT=MT*CT*T2
      A(I,7)=(QT/QS-1.0)*100.0
      T3=A(I,1)-A(I,4)
      T4=A(I,2)-A(I,3)
      IF(T3.EQ.T4)THEN
      T5=T3
      ELSE
      T5=(T3-T4)/ALOG(T3/T4)
      END IF
      IF(K.LT.1.5)THEN
      IF(T1.GE.T2)THEN
      Q=QS
      A(I,11)=(1.0420635*(1.0+0.1/T1)-1.0)*100.0
      B(I)='SHELL'
      ELSE
      Q=QT
```

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```
      A(I,11)=(1.0420635*(1.0+0.1/T2)-1.0)*100.0
      B(I)='TUBE'
      END IF
      ELSE
      Y=(1.0/T2)/(0.1394+1.01394/T1)
      IF(Y.LE.1.0)THEN
      Q=QT
      A(I,11)=(1.0420635*(1.0+0.1/T2)-1.0)*100.0
      B(I)='TUBE'
      ELSE
      Q=QS
      A(I,11)=(1.0565863*(1.0+0.1/T1)-1.0)*100.0
      B(I)='SHELL'
      END IF
      END IF
      A(I,9)=Q/(0.18063*T5)
      A(I,10)=A(I,9)*176.1
      A(I,8)=Q
      IF(I.EQ.K1)GOTO 40
      GOTO 30
   40 PRINT 50
   50 FORMAT(6X,21HTEMPERATURES  (DEG C),7X,10HFLOW RATES,6X,
     &4HHEAT,6X,11HDISSIPATION,6X,21HOVERALL HEAT TRANSFER,
     &5X,7HMAXIMUM)
      PRINT 60
   60 FORMAT(3X,9HSHELLSIDE,7X,8HTUBESIDE,6X,12HSHELL   TUBE,
     &4X,15HBALANCE   BASED,4X,5HVALUE,9X,11HCOEFFICIENT,8X,
     &11HUNCERTAINTY)
      PRINT 70
   70 FORMAT(2X,2(2HIN,6X,3HOUT,5X),2X,6H(KG/S),8X,5H(PCT),
     &5X,2HON,6X,4H(KW),10X,1H2,10X,4H2   0,5X,9H(+&- PCT))
      PRINT 80
   80 FORMAT(77X,8H(KW/M K),1X,13H(BTU/FT HR F))
      DO 100 I=1,K1
      PRINT 90,(A(I,J),J=1,7),B(I),(A(I,J),J=8,11)
   90 FORMAT (1X,4(F5.2,3X),2(F5.3,3X),1X,F5.1,4X,A5,3X,
     &2(F6.3,5X),F6.0,9X,F4.1)
  100 CONTINUE
      STOP
      END
      SUBROUTINE PROP(T,C)
      C=4.216188-2.401508E-3*T+4.362938E-5*T**2-2.027721E-7*T**3
      RETURN
      END
```

A3.2 PROGRAMME 'HEAT'

STEADY STATE HEAT TRANSFER ANALYSIS OF TEST RESULTS FOR TEST HEAT EXCHANGER
WATER/WATER DATA INPUT 1;OIL/WATER DATA INPUT 2 ?1

TEMPERATURES (DEG C)				FLOW RATES		HEAT	DISSIPATION		OVERALL HEAT TRANSFER COEFFICIENT		MAXIMUM
SHELLSIDE		TUBESIDE		SHELL	TUBE	BALANCE	BASED	VALUE			UNCERTAINTY
IN	OUT	IN	OUT	(KG/S)		(PCT)	ON	(KW)	$(KW/M^2 K)$	$(BTU/FT^2 HR\ ^0F)$	(+&- PCT)
50.58	43.97	19.95	26.47	0.149	0.151	0.4	SHELL	4.106	0.945	166.	5.8
49.48	43.32	20.20	26.17	0.151	0.151	-3.0	SHELL	3.891	0.928	163.	5.9
49.92	46.56	20.95	27.42	0.302	0.155	-1.3	TUBE	4.191	0.966	170.	5.8
49.16	47.09	20.40	27.57	0.529	0.151	-1.0	TUBE	4.532	1.043	184.	5.7
50.28	48.22	19.90	27.22	0.529	0.151	1.6	TUBE	4.627	1.001	176.	5.6
50.38	48.74	20.40	28.01	0.756	0.151	-7.2	TUBE	4.810	1.055	186.	5.6
49.83	48.37	19.65	27.12	0.756	0.151	2.4	TUBE	4.722	1.021	180.	5.6
49.37	40.56	20.30	24.03	0.151	0.340	-4.6	SHELL	5.564	1.357	239.	5.4
49.58	47.27	20.20	25.54	0.756	0.340	4.1	TUBE	7.595	1.647	290.	6.2
50.13	37.70	20.35	22.89	0.155	0.756	-0.2	SHELL	8.046	2.032	358.	5.0
49.32	41.46	20.45	23.60	0.302	0.756	0.3	SHELL	9.929	2.361	416.	5.5
49.78	44.48	19.95	23.70	0.529	0.756	1.2	SHELL	11.718	2.564	452.	6.2
49.48	45.48	20.00	24.00	0.756	0.756	0.1	SHELL	12.634	2.745	483.	6.8
49.37	33.68	20.55	22.13	0.151	1.512	0.8	SHELL	9.908	2.837	500.	4.9
50.08	38.90	20.95	23.03	0.302	1.512	-6.9	SHELL	14.122	3.523	620.	5.1
50.18	42.06	20.40	23.03	0.529	1.562	-4.3	SHELL	17.951	4.090	720.	5.5
49.47	43.46	20.30	23.33	0.756	1.512	0.9	SHELL	18.981	4.268	752.	5.9
50.13	31.23	19.85	20.88	0.151	2.646	-4.5	SHELL	11.934	3.490	615.	4.8
49.88	31.91	20.45	21.43	0.151	2.646	-4.4	SHELL	11.347	3.362	592.	4.8
50.33	36.40	20.05	21.63	0.302	2.646	-0.6	SHELL	17.594	4.438	782.	5.0
50.08	39.61	19.85	21.88	0.529	2.646	-2.9	SHELL	23.144	5.399	951.	5.2
50.18	40.11	20.70	22.68	0.529	2.646	-1.6	SHELL	22.261	5.307	935.	5.2
49.68	41.66	20.50	22.73	0.756	2.646	-2.6	SHELL	25.328	5.858	1032.	5.5
50.18	42.06	20.90	23.09	0.756	2.646	-5.5	SHELL	25.645	5.915	1042.	5.5

PROGRAM STOP AT 960

TABLE A3.1 FIRST WATER/WATER TEST RESULTS

STEADY STATE HEAT TRANSFER ANALYSIS OF TEST RESULTS FOR TEST HEAT EXCHANGER
WATER/WATER DATA INPUT 1;OIL/WATER DATA INPUT 2 ?1

TEMPERATURES (DEG C)				FLOW RATES		HEAT	DISSIPATION		OVERALL HEAT TRANSFER COEFFICIENT		MAXIMUM
SHELLSIDE		TUBESIDE		SHELL	TUBE	BALANCE	BASED	VALUE			UNCERTAINTY
IN	OUT	IN	OUT	(KG/S)		(PCT)	ON	(KW)	(KW/M^2 K)	(BTU/FT^2 HR $^\circ$F)	(+&- PCT)
49.94	43.36	19.47	25.81	0.151	0.151	-3.6	SHELL	4.156	0.958	169.	5.8
49.78	46.36	20.75	27.31	0.302	0.151	-4.1	TUBE	4.146	0.956	168.	5.8
49.78	46.26	20.75	27.31	0.302	0.151	-6.8	TUBE	4.146	0.958	169.	5.8
50.03	47.86	19.71	27.51	0.529	0.145	-1.5	TUBE	4.725	1.037	183.	5.5
50.08	47.91	19.76	27.56	0.529	0.145	-1.5	TUBE	4.725	1.037	183.	5.5
49.58	47.96	19.42	27.51	0.762	0.151	-0.9	TUBE	5.114	1.125	198.	5.5
49.18	40.68	19.42	23.12	0.151	0.340	-1.9	SHELL	5.369	1.261	222.	5.4
51.07	45.76	19.91	24.51	0.302	0.340	-2.5	SHELL	6.709	1.418	250.	6.2
50.97	45.66	19.81	24.41	0.302	0.340	-2.5	SHELL	6.709	1.418	250.	6.2
49.17	46.16	20.50	25.31	0.535	0.334	-0.3	TUBE	6.715	1.502	265.	6.4
50.28	47.96	21.00	26.11	0.768	0.334	-4.3	TUBE	7.133	1.546	272.	6.2
50.33	48.01	21.00	26.11	0.768	0.334	-4.3	TUBE	7.133	1.543	272.	6.2
49.78	38.02	19.76	22.07	0.151	0.756	-1.7	SHELL	7.427	1.815	320.	5.1
49.68	37.92	19.66	21.97	0.151	0.756	-1.7	SHELL	7.427	1.815	320.	5.1
50.77	42.96	20.21	23.32	0.305	0.756	-1.4	SHELL	9.970	2.205	388.	5.5
50.72	42.91	20.21	23.32	0.305	0.756	-1.4	SHELL	9.970	2.210	389.	5.5
50.83	45.48	20.21	23.81	0.529	0.756	-3.8	SHELL	11.830	2.506	441.	6.2
49.93	45.58	19.61	23.78	0.756	0.756	-4.1	SHELL	13.740	2.919	514.	6.6
50.78	35.56	20.60	22.12	0.151	1.512	-0.0	SHELL	9.612	2.525	445.	4.9
49.47	39.42	20.95	22.97	0.302	1.512	0.6	SHELL	12.695	3.159	556.	5.2
50.57	42.96	20.65	23.32	0.529	1.512	0.3	SHELL	16.825	3.771	664.	5.6
50.17	44.16	20.40	23.32	0.756	1.512	-2.7	SHELL	18.983	4.158	732.	5.9
49.38	31.46	19.52	20.53	0.151	2.646	-1.2	SHELL	11.315	3.268	576.	4.8
49.57	37.47	19.72	21.22	0.318	2.646	3.2	SHELL	16.079	3.932	692.	5.1
50.82	40.51	19.61	21.63	0.529	2.646	-1.9	SHELL	22.792	5.085	895.	5.2
50.67	42.81	20.16	22.42	0.756	2.646	0.7	SHELL	24.825	5.422	955.	5.5
50.67	37.87	19.52	22.02	0.151	0.756	-2.2	SHELL	8.084	1.936	341.	5.0
51.46	47.26	20.21	24.31	0.756	0.756	-2.3	SHELL	13.268	2.711	477.	6.7
50.97	34.86	20.85	22.42	0.151	1.512	-2.4	SHELL	10.174	2.758	486.	4.9

PROGRAM STOP AT 960

TABLE A3.2 SECOND WATER WATER TEST RESULTS

STEADY STATE HEAT TRANSFER ANALYSIS OF TEST RESULTS FOR TEST HEAT EXCHANGER
WATER/WATER DATA INPUT 1;OIL/WATER DATA INPUT 2 ?2

TEMPERATURES (DEG C) SHELLSIDE IN	SHELLSIDE OUT	TUBESIDE IN	TUBESIDE OUT	FLOW RATES (KG/S) SHELL	FLOW RATES (KG/S) TUBE	HEAT BALANCE (PCT)	DISSIPATION BASED ON	DISSIPATION VALUE (KW)	OVERALL HEAT TRANSFER COEFFICIENT (KW/M 2 K)	OVERALL HEAT TRANSFER COEFFICIENT (BTU/FT 2 HR 0 F)	MAXIMUM UNCERTAINTY (+&- PCT)
49.56	43.36	20.94	23.85	0.155	0.151	-2.7	SHELL	1.891	0.436	77.	7.4
49.95	45.80	20.15	24.05	0.302	0.151	-0.6	TUBE	2.466	0.530	93.	6.9
49.25	46.40	19.39	24.00	0.531	0.151	-2.7	TUBE	2.915	0.618	109.	6.5
49.85	47.66	19.74	24.90	0.761	0.151	-1.2	TUBE	3.262	0.684	120.	6.2
50.36	42.99	20.75	22.29	0.154	0.340	-2.2	SHELL	2.240	0.495	87.	7.1
49.25	44.44	20.40	22.39	0.299	0.340	-0.5	SHELL	2.844	0.619	109.	7.9
49.50	46.15	20.70	23.19	0.540	0.340	-1.1	TUBE	3.542	0.758	133.	8.4
49.40	46.76	20.85	23.60	0.763	0.340	-1.9	TUBE	3.912	0.838	148.	8.0
49.30	41.28	20.45	21.17	0.156	0.756	-7.5	SHELL	2.461	0.561	99.	7.0
49.71	43.89	20.65	21.68	0.298	0.756	-4.8	SHELL	3.420	0.741	130.	7.5
50.06	45.75	20.69	22.09	0.533	0.756	-2.6	SHELL	4.542	0.949	167.	8.1
49.30	45.85	20.40	21.99	0.763	0.756	-3.4	SHELL	5.202	1.092	192.	8.7
49.76	41.64	20.56	21.33	0.153	0.756	-0.8	SHELL	2.453	0.553	97.	7.0
49.66	40.78	20.10	20.47	0.153	1.512	-12.7	SHELL	2.680	0.601	106.	6.8
50.36	43.74	20.50	21.07	0.296	1.512	-6.7	SHELL	3.865	0.818	144.	7.3
49.76	44.85	20.65	21.44	0.530	1.512	-2.9	SHELL	5.143	1.086	191.	7.8
49.45	45.51	20.95	21.84	0.765	1.512	-5.6	SHELL	5.959	1.266	223.	8.3
50.15	45.75	20.15	21.53	0.522	0.756	-3.9	SHELL	4.542	0.928	164.	8.1
49.76	46.30	20.45	22.04	0.759	0.756	-3.3	SHELL	5.199	1.075	189.	8.7
49.05	45.15	20.70	21.59	0.766	1.512	-4.6	SHELL	5.900	1.260	222.	8.4
50.36	41.48	20.45	20.82	0.155	1.512	-13.9	SHELL	2.717	0.601	106.	6.8

PROGRAM STOP AT 960

TABLE A3.3 OIL/WATER TEST RESULTS

A3.4. SPECIMEN CALCULATIONS

FIRST WATER/WATER TESTS, FIRST RUN

$$\dot{m}_h = 0.1486 \text{ kg/s}$$

$$\theta_{h1} = 50.58 \ ^oC$$

$$\theta_{h2} = 43.97 \ ^oC$$

$$\dot{m}_c = 0.1512 \text{ kg/s}$$

$$\theta_{c1} = 19.95 \ ^oC$$

$$\theta_{c2} = 26.47 \ ^oC$$

Shellside mean temperature $= \dfrac{50.58 + 43.97}{2}$

$= \underline{47.27 \ ^oC}$

$c_h = 4.216188 - 2.401508 \times 10^{-3} \times 47.27 + 4.362938 \times 10^{-5} \times 47.27^2 - 2.027721 \times 10^{-7} \times 47.27^3$ kJ/kgK

$\therefore \underline{c_h = 4.1787 \text{ kJ/kgK}}$

tubeside mean temperature $= \dfrac{19.95 + 26.47}{2}$

$= \underline{23.21 \ ^oC}$

similarly, $\underline{c_c = 4.1814 \text{ kJ/kgK}}$

$\Delta\theta_h = 50.58 - 43.97 = \underline{6.61K}$

$\dot{Q}_h = 0.1486 \dfrac{kg}{s} \times 4.1787 \dfrac{kJ}{kgK} \times 6.61K = \underline{4.1045 \text{ kW}}$

$\Delta\theta_c = 26.47 - 19.95 = \underline{6.52 \text{ K}}$

$\dot{Q}_c = 0.1512 \times 4.184 \times 6.52 = \underline{4.1221 \text{ kW}}$

heat balance $= \left(\dfrac{4.1221}{4.1045} - 1\right) \times 100 = \underline{0.4\%}$

$\Delta\theta_h > \Delta\theta_c \therefore$ the shellside dissipation rate has the smaller uncertainty.

$\therefore$ estimated maximum uncertainty in the overall heat transfer coefficient U =

$$\pm\ 1.0420635 \left(1 - \frac{0.1}{6.61}\right) \times 100 = \pm\ \underline{5.8\ \%}$$

$\Delta\theta_{LMTD}$ (counter flow) =

$$\frac{(50.58 - 26.47) - (43.97 - 19.95)}{\ln\left(\frac{50.58 - 26.47}{43.97 - 19.95}\right)}$$

$$= \underline{24.065\ K}$$

$\therefore$ U (based on shellside dissipation)

$$= \frac{4.1045\ kW}{0.18063\,m^2 \times 24.065\,K}$$

$$= \underline{0.944\ kW/m^2K}$$

A3.5. FOULING ASSESSMENT

The purpose of this fouling assessment is to give an indication of the magnitude of the drop in performance of the test unit if an allowance for fouling on both the tube inside and outside surfaces is made.

EQUATIONS

$$\frac{1}{U_{clean}} = \frac{1}{h_o} + \frac{1}{h_{ir}} \qquad (A3.14)$$

$$\frac{1}{U_{fouled}} = \frac{1}{h_o} + \frac{1}{h_{ir}} + r_{fo} + r_{fi}\left(\frac{d_o}{d_i}\right) \qquad (A3.15)$$

where U_{clean} = the overall heat transfer coefficient for a unfouled heat exchanger

U_{fouled} = the overall heat transfer coefficient for a heat exchanger subjecting to fouling

DATA

The partial heat transfer coefficients were calculated from standard Serck data. A summary is given below:

Tubeside partial referred to outside surface

Fluid: water

Flow rate	h_{ir}	
(kg/s)	(kW/m^2K)	($Btu/ft^2h\ R$)
0.151	0.920	162
1.512	4.776	841
2.646	7.564	1332

Shellside partials

Fluid: water

Flow rate	h_o	
(kg/s)	(kW/m^2K)	($Btu/ft^2h\ R$)
0.151	4.463	786
0.756	10.329	1819

Fluid: oil (Shell Diala B)

Flow rate	h_o	
(kg/s)	(kW/m^2K)	($Btu/ft^2h\ R$)
0.151	0.874	154
0.756	1.732	305

Fouling factors were taken from reference 15 and are:

mains water : 0.001 $ft^2h\ R/Btu = 0.1761\ m^2K/kW$

transformer oil: 0.001 $ft^2h\ R/Btu = 0.1761\ m^2K/kW$

Substituting these values into the above equation for U_{fouled} and with $d_o = 6.25$ mm and $d_i = 5.55$ mm:

$$\frac{1}{U_{fouled}} = \frac{1}{h_o} + \frac{1}{h_{ir}} + 0.3744 \qquad m^2K/kW$$

The summary of the calculations for the water/water and oil/water cases at the extremes of the flow ranges is given below in tables A3.4 and A3.5.

TABLE A3.4. THE EFFECT OF FOULING ON THE PERFORMANCE OF THE TEST UNIT - WATER/WATER

SHELL FLOW RATE (kg/s)		TUBE FLOW RATE (kg/s)		
		0.151	1.512	2.646
0.151	U_{clean} (kW/m^2K)	0.761	2.306	2.805
	U_{fouled} (kW/m^2K)	0.591	1.232	1.363
	% age drop	22	46	51
0.756	U_{clean} (kW/m^2K)	0.846	3.265	4.367
	U_{fouled} (kW/m^2K)	0.642	1.465	1.652
	% age drop	24	55	62

TABLE A3.5. THE EFFECT OF FOULING ON THE PERFORMANCE OF THE TEST UNIT - OIL/WATER

SHELL OIL FLOW RATE (kg/s)		TUBE FLOW RATE (kg/s)	
		0.151	1.512
0.151	U_{clean} (kW/m^2K)	0.448	0.739
	U_{fouled} (kW/m^2K)	0.384	0.579
	% age drop	14	22
0.756	U_{clean} (kW/m^2K)	0.601	1.271
	U_{fouled} (kW/m^2K)	0.490	0.861
	% age drop	18	32

APPENDIX A4

DYNAMIC TESTS

A4.1. THERMOPILE CALIBRATION EQUATIONS

The thermopile calibration equations are of the form,

$$T = A + BV + CV^2 + DV^3 + EV^4 \pm 0.1K$$

where T = Temperature (^oC)

V = e.m.f. (mV)

A,B,C,D,E = constants

The value of the constants for each thermopile are given in the table below.

CONSTANTS	THERMOPILE LOCATION			
	SHELL INLET	SHELL OUTLET	TUBE INLET	TUBE OUTLET
A	1.631689 E-2	-2.144337 E-3	2.467203 E-2	3.481293 E-2
B	6.484883 E+0	6.509252 E+0	6.478119 E+0	6.453876 E+0
C	-6.534999 E-2	-7.196949 E-2	-6.245087 E-2	-5.432990 E-2
D	3.822533 E-3	4.328189 E-3	3.456007 E-3	2.452989 E-3
E	-1.485299 E-4	-1.588805 E-4	-1.329752 E-4	-9.468164 E-5

A4.2. DETAILS OF TEST EQUIPMENT

A4.2.1. WATER/WATER TEST RIG

(Equipment reference letters refer to those in fig. 4.4)

Hot tank:		1.050 m x 0.915 m x 0.590 m
Cold tank:		0.890 m x 0.890 m x 0.890 m
Tank insulation:		50 mm thick Rocksil
Pumps:	P1	ITT Jabsco, max capacity 16 litre/ min.
	P2	Worthington Simpson, 6 hp. centrifugal pump, size $1\frac{1}{2}$ DM8, No. 5162979
	P3	Worthington Simpson, centrifugal pump, model no. unknown

Orifice units, (including 2 inch dia. bore manual gate valves upstream and downstream of orifice):

Shellside circuit:

0.251 inch orifice in 2 inch dia pipe

0.505 inch orifice in 2 inch dia pipe

Tubeside circuit:

0.265 inch dia. orifice in 1.5 inch dia. pipe

0.505 inch dia. orifice in 2 inch dia. pipe

0.715 inch dia. orifice in 2 inch dia. pipe

Steam heaters H1, H2: Serck designation TSS-23, shell and tube type

Valves:

SV1, SV2	Alexander Controls solenoid valves, gate/piston type, 1 1/2" dia. bore
V1	1 inch manual globe valve
V2	2 inch manual gate valve
V3	$1\frac{1}{2}$ inch manual gate valve
V4-V10	1 inch manual gate valve
SFV1, SFV2	Steam flow regulator valves, comprising:- 1 inch manual gate valve; $\frac{1}{4}$ inch manual gate valve
CV1, CV2	Constant pressure reduction valves, 70-30 lbf/in^2

Pressure gauges, P; Bourden tube, 0 - 100 lbf/in^2

Differential pressure across orifices measured with 48 inch mercury 'U' tube manometer.

Pipework:

Shellside circuit (including shellside heating circuit)	- from tanks to just beyond shellside outlet temperature measuring point, 1 inch dia. ABS; remainder (except for orifice units) 1 inch dia. copper piping
Tubeside circuit	- from suction side of pump to valve V4, 2 inch dia copper piping - piping at tubeside inlet and outlet temperature measuring points, 1 inch ABS - remainder 1 inch dia. copper piping
Steam circuit	- 1 inch dia. copper piping from steam main.

A4.2.2. OIL/WATER TEST RIG

(Equipment reference letters refer to those in fig. 4.5)

Hot and Hot Dump Tanks:		0.915 m x 0.915 m x 0.915 m
Cold Tank:		0.890 m x 0.890 m x 0.890 m
Cold Dump Tank:		1.050 m x 0.915 m x 0.590 m
Tank Insulation:		50 mm thick Rocksil
Pumps:	P1	Gear pump, indentification unknown
	P2	Worthington Simpson, 6 h.p. centrifugal pump, size $1\frac{1}{2}$ DM8, No. 5162979
	P3	Worthington Simpson centrifugal pump, size $1\frac{1}{2}$, No. 972042
	P4	B.S.A. Gear pump, no other details available

Orifice units, (including 2 inch dia. bore manual gate valves upstream and downstream of orifice):

0.265 inch dia. orifice in 1.5 inch dia. pipe

0.505 inch dia. orifice in 2 inch dia. pipe

0.715 inch dia. orifice in 2 inch dia. pipe

Oil flow meter:	Rotameter Manufacturing Co., Serial No. R264439, Tube No. 233500/43083/T, 900 - 7500 lb/hr. in 100 lb/hr increments.
Steam Heaters H1, H2 and Oil Cooler C1:	Serck designation TSS-23, shell and tube type
'Hot & Cold' supply tank heaters, E1 and E2:	3 x 2kW electric immersion heaters
Valves: SV1, SV2, SV3, SV4	Dewraswitch Asco 2 way solenoid valves, 1 inch dia. pipe size, catalogue no. 8210B54

V1 - V3	2 inch manual gate valve
V4 - V9	$1\frac{1}{2}$ inch manual gate valve
V10-V17	1 inch manual gate valve
SFV1, SFV2	Steam flow regulator valves, comprising:- 1 inch manual gate valve $\frac{1}{4}$ inch manual gate valve
CV1, CV2	Constant pressure reduction valves, 70-30 lbf/in^2

Thermometer, TP: Class A, mercury in glass thermometer(s) in pocket

Thermocouples, TH: Chromel/Alumel thermocouples and CRL digital display

Pressure gauges P: Bourden Tube, 0 - 100 lbf/in^2

Differential pressure across orifices measured with 48 inch mercury 'U' tube manometer

Pipework:

Shellside circuit:	from supply tanks to just beyond shellside outlet temperature measuring point, 1 inch dia. ABS; remainder, 1 inch dia. copper piping
Tubeside circuit:	as for water/water test rig
'Hot' oil heating circuit:	1 inch dia copper piping
'Cold' oil heating circuit:	1 inch dia copper piping
Steam circuits:	1 inch dia. copper piping from steam main

A4.2.3. DATA LOGGING EQUIPMENT

Solarton Schlumberger Data Logger, comprising:-

A200 Ditigal Volt Meter (DVM)

Data Transfer Unit (DTU), with the following cards:

Scanner Controller 3215

Controller 3211

Clock 3210

DVM Interface 3205

Output Driver 3211 (for Facit Tape Punch)

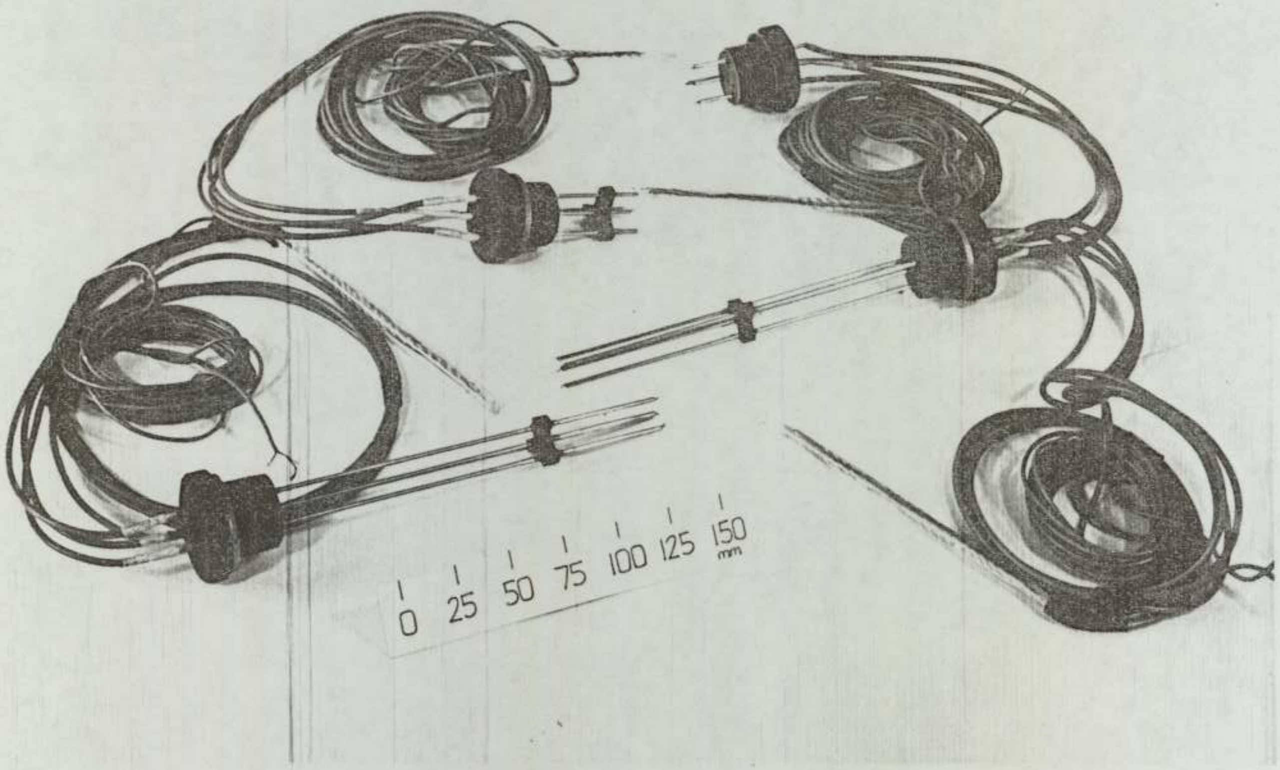
Output Driver 3224 (for Data Dynamics Teletype)

Power Supply

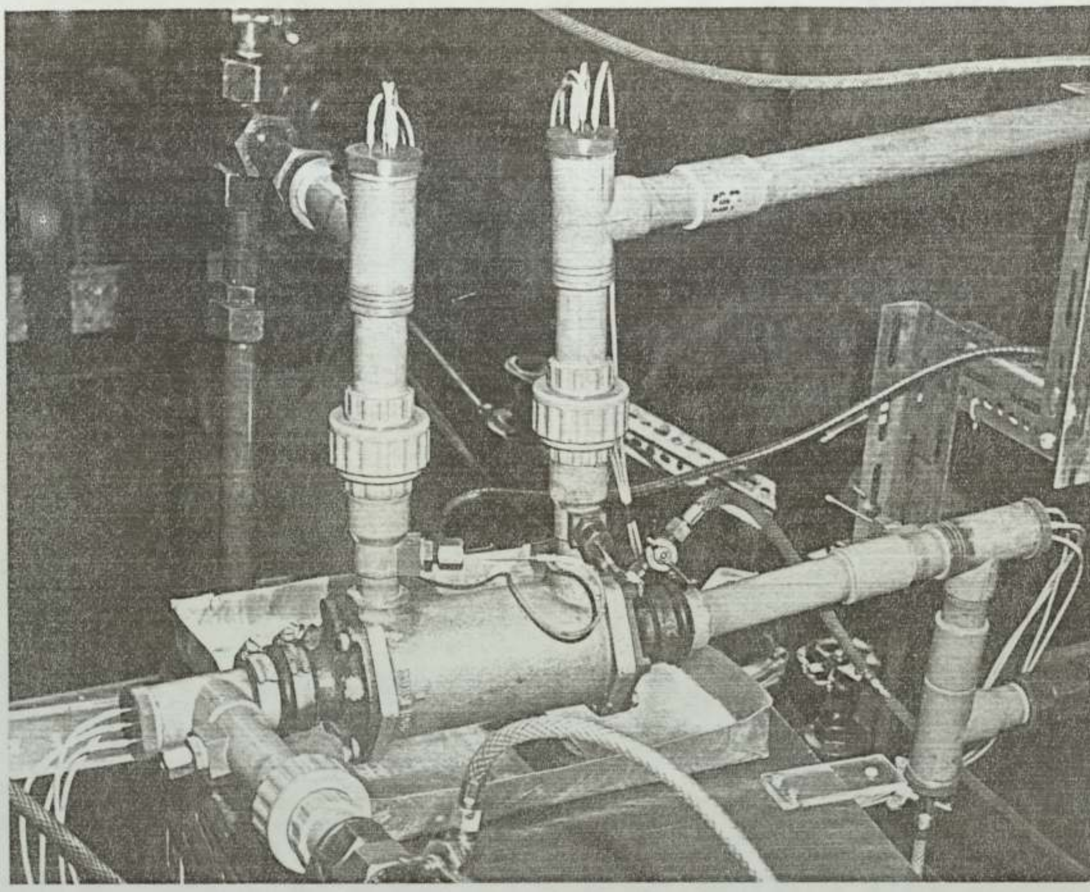
Analogue Scanner

Facit 4070 Tape Punch

Data Dynamics Teletype



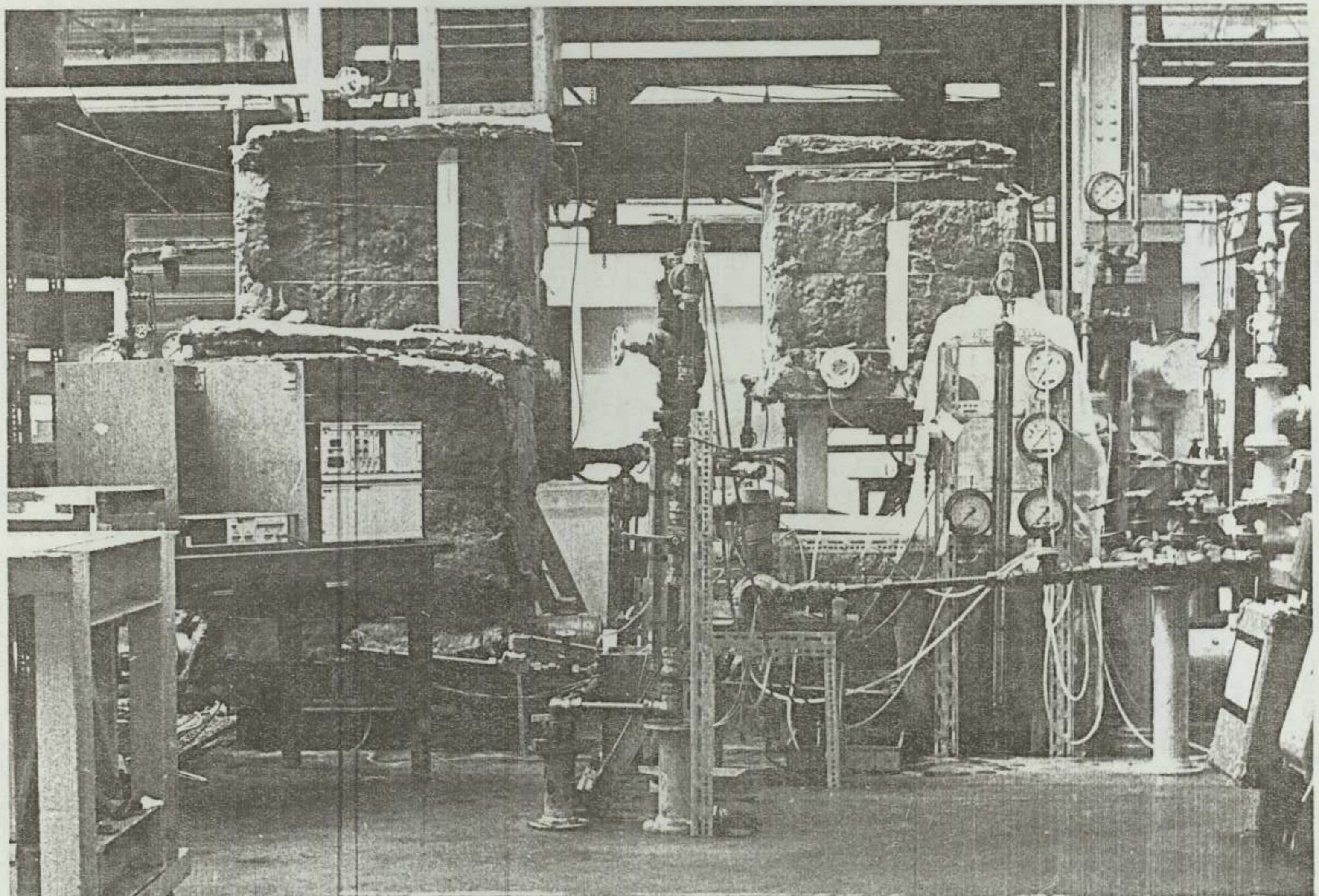
PHOTOGRAPH 2. THERMOPILES



PHOTOGRAPH 3. THE TEST UNIT ON THE DYNAMIC OIL/WATER TEST RIG.

DESCRIPTION OF PHOTOGRAPH 4:

REAR LEFT	'HOT' OIL SUPPLY TANK ON STAND
REAR RIGHT	'COLD' OIL SUPPLY TANK ON STAND
FRONT LEFT	DATA LOGGING EQUIPMENT
FRONT CENTRE	OIL FLOW METER (VERTICAL) & TEST UNIT (LAGGED)
FRONT RIGHT	WATER FLOW MANOMETER & ORIFICE UNITS (BEHIND CLUSTER OF VALVES)
FRONT FAR RIGHT	EDGE OF CONTROL PANEL



PHOTOGRAPH 4 GENERAL VIEW OF OIL/WATER DYNAMIC TEST RIG (DESCRIPTION ON PAGE 183)

APPENDIX A5

DYNAMIC TESTS: SPECIMEN PROGRAMME OUTPUT

. A5.1 SPECIMEN OUTPUT FROM PROGRAMME 'SORT' (RUN 7)

```
WATER/WATER DATA INPUT 1;OIL/WATER DATA INPUT 2 ?1

INPUT FILENAME ?RW7

INPUT SHELL FLOW,LB/HR ?1180

INPUT TIME INTERVAL,NO. OF ROWS IN INCREMENT ?1

+VE/-VE DATA: +VE STEP ONLY,INPUT 1;+VE & -VE STEP,INPUT 2;
-VE STEP ONLY,INPUT 3; +VE DATA ONLY: INPUT 4 ?1

IF START OF STEP OUTPUT REQUIRED INPUT 1,IF NOT 2 ?1
```

37	21.03	20.68	20.23	20.28
38	21.06	20.66	20.26	20.31
39	21.03	20.69	20.28	20.33
40	21.03	20.70	20.25	20.32
41	21.03	20.70	20.21	20.30
42	21.03	20.68	20.17	20.25
43	21.03	20.67	20.19	20.24
44	21.02	20.66	20.23	20.28
45	21.02	20.67	20.28	20.31
46	21.04	20.68	20.29	20.34
47	21.54	20.68	20.30	20.36
48	23.24	20.73	20.31	20.42
49	25.82	21.01	20.33	20.54
50	28.71	21.76	20.33	20.72
51	31.55	22.89	20.34	20.94
52	34.16	24.29	20.34	21.17
53	36.21	25.76	20.30	21.35
54	38.14	27.10	20.25	21.49
55	39.56	28.31	20.22	21.62
56	40.83	29.27	20.20	21.73
57	41.89	30.19	20.19	21.83
150	49.92	36.98	20.26	22.80
151	49.94	37.01	20.31	22.85
152	49.92	37.07	20.31	22.86
153	49.94	37.03	20.32	22.88
154	49.95	37.05	20.33	22.86
155	49.94	37.04	20.32	22.88
156	49.95	37.13	20.26	22.85
157	49.95	37.02	20.23	22.80
158	49.95	36.98	20.25	22.79
159	49.95	37.05	20.28	22.83
160	49.95	36.99	20.29	22.85
161	49.95	37.11	20.29	22.85
162	49.97	37.03	20.28	22.83
163	49.95	37.07	20.23	22.82
164	49.98	36.99	20.20	22.78
165	49.93	37.02	20.25	22.79
166	49.98	36.99	20.25	22.79
167	49.98	37.06	20.24	22.82
168	49.99	37.01	20.26	22.82
169	49.99	37.07	20.26	22.84
170	49.99	37.04	20.26	22.85
171	50.00	37.19	20.26	22.85
172	49.99	37.16	20.29	22.87
173	50.00	37.20	20.29	22.88
174	49.99	37.16	20.29	22.85
175	49.99	37.20	20.30	22.86

```
TEMP DIFF PARAMETERS ARE: 0.04,0.20,0.20
NEW TEMP DIFF PARAMETERS REQUIRED? NO 1; YES 2 ?2

RE-LOCATE START OF SHELL INLET RESPONSE? NO 1; YES 2 ?1

NEW SHELL OUTLET TEMP DIFF= ?0.2

SHELL OUTLET RESPONSE INDICATED IN ROW: ?49

NEW TUBE OUTLET TEMP DIFF= ?0.1

TUBE OUTLET RESPONSE INDICATED IN ROW: ?49

DYNAMIC TEST RUN NO.-RW7
DELAY BEFORE TEMP. STEP ARRIVES AT SHELL INLET   8.96 SE
SHELL OUTLET LAG   2.70 SEC
TUBE OUTLET LAG   3.30 SEC
```

CONTINUED

A5.1 SPECIMEN OUTPUT FROM PROGRAMME 'SORT' (RUN 7) CONTINUED

```
COPY OF TIME/TEMP DATA REQUIRED? ALL 1; PART 2; NONE 3?2

TIME IN SEC;  TEMP IN DEG C
 TIME    TH1      TIME    TH2        TIME    TC1        TIME    TC2
  0.0   21.04      0.0   20.73        0.0   20.29        0.0   20.42
  1.2   21.54      1.2   21.01        1.2   20.30        1.2   20.54
  2.4   23.24      2.4   21.70        2.4   20.31        2.4   20.72
  3.6   25.82      3.6   22.89        3.6   20.33        3.6   20.94
  4.8   28.71      4.8   24.29        4.8   20.33        4.8   21.17
  6.0   31.55      6.0   25.70        6.0   20.34        6.0   21.35
  7.2   34.16      7.2   27.11        7.2   20.34        7.2   21.49
  8.4   36.21      8.4   28.31        8.4   20.26        8.4   21.62
  9.6   38.14      9.6   29.27        9.6   20.25        9.6   21.73
 10.8   39.56     10.8   30.19       10.8   20.22       10.8   21.83

INITAL TEMPERATURES; DEG C.
 TH1    TH2    TC1    TC2
21.04  20.73  20.29  20.42

FINAL TEMPERATURES; DEG C.
 TH1    TH2    TC1    TC2
4[illegible].94  37.04  20.32  22.[illegible]

MAGNITUDE OF STEP INPUT=  20.90  DEG C

AVERAGE TUBE INLET TEMPERATURE= 20.28  DEG C
MIN= 20.13  MAX= 20.41
STANDARD DEVIATION OF TUBE INLET TEMPERATURE=    5.12455 9E-02

NO. OF TEMPERATURE SAMPLES:
  TH1=          110
  TH2=          108
  TC1=          110
  TC2=          108

21.04
21.

PAUSED,FEED BLANK TAPE

20.73
21.01

PAUSED,FEED BLANK TAPE

20.29
20

PAUSED,FEED BLANK TAPE

20.42
20.54

PAUSED,FEED BLANK TAPE

PROGRAM STOP AT    2520
```

SHELL INLET			SHELL OUTLET			TUBE OUTLET		
FREQUENCY (rad/s)	AMPLITUDE (dbs)	PHASE ANGLE (°)	FREQUENCY (rad/s)	AMPLITUDE (dbs)	PHASE ANGLE (°)	FREQUENCY (rad/s)	AMPLITUDE (dbs)	PHASE ANGLE (°)
.00200	52.62903	-2.23857	.00200	78.05150	-199.99942	.00200	40.94151	-2.91753
.00400	52.60706	-4.47105	.00400	65.09515	-224.86626	.00400	40.91278	-5.81639
.00600	52.57274	-6.69292	.00600	57.54720	-259.20923	.00600	40.84137	-8.67752
.00800	52.51401	-8.89797	.00800	54.10777	62.88141	.00800	40.74419	-11.48682
.01000	52.43779	-11.07798	.01000	53.27991	34.71590	.01000	40.61852	-14.22529
.02000	51.79309	-21.26595	.02000	52.18881	-19.74904	.02000	39.62631	-26.24770
.03000	50.75235	-29.03203	.03000	49.47057	-36.45010	.03000	38.25516	-33.99529
.04000	49.59915	-32.73511	.04000	47.36508	-34.16443	.04000	37.01027	-36.90359
.05000	48.84395	-33.10289	.05000	47.35675	-31.17311	.05000	36.30739	-37.65828
.06000	48.62431	-35.89711	.06000	47.46530	-35.99158	.06000	35.98901	-39.60283
.07000	48.51541	-37.02838	.07000	46.99326	-41.33551	.07000	35.67903	-43.04330
.08000	48.21532	-41.11432	.08000	46.35471	-43.92657	.08000	35.26379	-46.73654
.09000	47.77783	-44.65787	.09000	46.06059	-46.61632	.09000	34.81050	-50.16872
.10000	47.33417	-47.61528	.10000	45.66060	-50.79219	.10000	34.33597	-53.55347
.15000	45.17300	-56.94719	.15000	43.22374	-59.99438	.15000	31.60478	-62.96977
.20000	43.68541	-69.44800	.20000	41.55934	-71.20067	.20000	30.09335	-73.36250
.25000	41.92014	-72.13608	.25000	39.83113	-74.62680	.25000	28.06088	-76.94370
.30000	40.65211	-78.86244	.30000	38.36259	-79.84625	.30000	26.43883	-81.91650
.35000	39.29309	-80.13049	.35000	37.03900	-81.27632	.35000	25.32831	-83.13309
.40000	37.97054	-83.96725	.40000	35.57922	-84.93259	.40000	24.19472	-85.48457
.45000	37.01904	-86.30219	.45000	34.66789	-86.90547	.45000	22.88345	-86.34819
.50000	35.85192	-87.89273	.50000	33.58414	-87.43741	.50000	21.51260	-89.68967
1.00000	28.64896	-91.57448	1.00000	26.56984	-90.87204	1.00000	16.50172	-93.36636
1.50000	25.08630	-90.39023	1.50000	23.21986	-90.72772	1.50000	12.02954	-86.02701
2.00000	22.47668	-89.12213	2.00000	20.73544	-90.45551	2.00000	8.15874	-91.00865
2.50000	20.74827	-89.95567	2.50000	18.64503	-90.00812	2.50000	6.68649	-86.24600
3.00000	19.06384	-89.65469	3.00000	17.05301	-90.39784	3.00000	4.85593	-89.27631
4.00000	16.55048	-90.25151	4.00000	14.51136	-89.86894	4.00000	3.16337	-90.30746
6.00000	13.07154	-90.88252	6.00000	11.11539	-90.22804	6.00000	-.42721	-89.45776
8.00000	10.47638	-89.90462	8.00000	8.54896	-89.85248	8.00000	-2.98114	-90.84810
10.00000	8.50867	-89.87004	10.00000	6.58800	-89.83017	10.00000	-4.92685	-89.78323
20.00000	2.55049	-90.07767	20.00000	.59295	-89.97696	20.00000	-10.90455	-90.16011
40.00000	-3.45899	-90.03593	40.00000	-5.39930	-89.94666	40.00000	-16.95377	-89.91425
60.00000	-6.98716	-89.96523	60.00000	-8.92406	-90.00461	60.00000	-20.46693	-90.09375
80.00000	-9.48484	-90.00050	80.00000	-11.42132	-89.98466	80.00000	-22.95818	-89.97090
100.00000	-11.41769	-89.99002	100.00000	-13.35932	-89.99281	100.00000	-24.39586	-89.97276

A5.2.1 TEMPERATURE ERROR SIGNAL FREQUENCY RESPONSE DATA FOR WATER/WATER RUN NO. 1, (PROGRAMME 'DFTR' OUTPUT)

SHELL INLET			SHELL OUTLET			TUBE OUTLET		
FREQUENCY (rad/s)	AMPLITUDE (dbs)	PHASE ANGLE (°)	FREQUENCY (rad/s)	AMPLITUDE (dbs)	PHASE ANGLE (°)	FREQUENCY (rad/s)	AMPLITUDE (dbs)	PHASE ANGLE (°)
.00200	64.72707	-197.80070	.00200	65.40741	-196.27597	.00200	31.56720	-2.60053
.00400	47.97504	-250.98938	.00400	50.40306	-226.81944	.00400	31.52634	-5.18410
.00600	47.63315	32.34373	.00600	43.63063	66.82068	.00600	31.46721	-7.73672
.00800	49.72346	12.62150	.00800	44.97241	27.88171	.00800	31.37901	-10.24154
.01000	50.61305	4.07142	.01000	46.15002	12.72188	.01000	31.26649	-12.68373
.02000	51.13828	-14.45737	.02000	47.12036	-13.13768	.02000	30.38289	-23.34283
.03000	50.36906	-23.85780	.03000	46.31520	-24.43245	.03000	29.17589	-30.10271
.04000	49.45192	-28.86411	.04000	45.29240	-30.03101	.04000	28.11737	-32.49728
.05000	48.75999	-31.32045	.05000	44.55333	-32.38731	.05000	27.60341	-33.15430
.06000	48.34518	-33.46286	.06000	44.17978	-34.52559	.06000	27.47119	-35.55084
.07000	48.01721	-36.35844	.07000	43.90465	-37.93253	.07000	27.27550	-40.19766
.08000	47.61829	-39.55064	.08000	43.49673	-41.93959	.08000	26.78973	-45.41136
.09000	47.15402	-42.21837	.09000	42.94872	-45.26736	.09000	26.06671	-49.44778
.10000	46.72511	-44.17799	.10000	42.41028	-47.39423	.10000	25.31177	-51.59140
.15000	45.00386	-54.00415	.15000	40.58430	-57.79927	.15000	23.73935	-59.24640
.20000	43.46562	-60.95258	.20000	38.80420	-64.64322	.20000	20.72185	-60.39365
.25000	42.07607	-66.93646	.25000	37.29297	-71.09790	.25000	20.83359	-68.25250
.30000	40.85793	-71.77446	.30000	35.99506	-75.78487	.30000	18.72737	-76.36188
.35000	39.63948	-75.88623	.35000	34.62921	-78.50421	.35000	16.85089	-71.18549
.40000	38.48007	-79.65010	.40000	33.35316	-82.32480	.40000	15.05262	-80.17345
.45000	37.40239	-81.71553	.45000	32.16548	-83.86513	.45000	12.89468	-87.69814
.50000	36.44063	-83.71181	.50000	31.24776	-86.02553	.50000	14.76431	-85.67085
1.00000	29.58080	-90.04510	1.00000	24.44867	-90.93538	1.00000	7.36376	-87.90952
1.50000	25.76855	-90.61629	1.50000	21.01595	-91.45900	1.50000	4.75317	-91.33963
2.00000	23.13046	-90.54953	2.00000	18.34111	-91.39146	2.00000	1.14277	-92.22687
2.50000	21.27769	-90.20651	2.50000	16.54338	-90.15973	2.50000	-.03233	-90.72501
3.00000	19.70862	-89.51497	3.00000	14.72059	-90.70955	3.00000	-2.18832	-91.82285
4.00000	17.10773	-90.10013	4.00000	12.13161	-88.89607	4.00000	-4.35789	-88.67241
6.00000	13.74093	-89.92746	6.00000	8.72598	-89.93469	6.00000	-7.61699	-90.56134
8.00000	11.14075	-90.18164	8.00000	6.25307	-89.41379	8.00000	-10.30798	-88.99158
10.00000	9.15854	-89.60060	10.00000	4.19342	-89.69301	10.00000	-12.14913	-88.98030
20.00000	3.17917	-90.00290	20.00000	-1.78157	-90.02226	20.00000	-18.19554	-89.70575
40.00000	-2.82629	-90.03493	40.00000	-7.81233	-89.98311	40.00000	-24.18346	-90.10168
60.00000	-6.34023	-89.99615	60.00000	-11.30372	-90.02483	60.00000	-27.75157	-90.16222
80.00000	-8.84114	-90.00050	80.00000	-13.80771	-89.98193	80.00000	-30.23599	-89.90337
100.00000	-10.77573	-89.96585	100.00000	-15.74598	-89.97580	100.00000	-32.18427	-89.98730

A5.2.2 TEMPERATURE ERROR SIGNAL FREQUENCY RESPONSE DATA FOR WATER/WATER RUN NO. 7, (PROGRAMME 'DFTR' OUTPUT)

A5.3 SPECIMEN OUTPUT FROM PROGRAMME 'FITFRØ (RUN 14)

```
 RUN NUMBER: 14TH1 -10

DATA FOR: SHELL INLET

    SUM OF SQUARES OF ERROR OF FIT =   5.2538E 02

.....0-N...........DENOM COEFFS.....
     0                          1.0000E 00
     1                          3.3849E 00

....0-N............NUMER COEFFS.....
     0                          1.0139E 02

......REAL POLE.......IMAG POLE.....
     -2.9549E-01         0.0000E 00

GAIN=       29.95763

STEADY STATE GAIN=       101.39417

 RUN NUMBER: 14TH2 -10

DATA FOR: SHELL OUTLET

    SUM OF SQUARES OF ERROR OF FIT =   2.8768E 02

.....0-N...........DENOM COEFFS.....
     0                          1.0000E 00
     1                          3.7092E 00

....0-N............NUMER COEFFS.....
     0                          1.1292E 02

......REAL POLE.......IMAG POLE.....
     -2.6960E-01         0.0000E 00

GAIN=       30.44359

STEADY STATE GAIN=       112.92160

 RUN NUMBER: 14TC2 -10

DATA FOR: TUBE OUTLET

    SUM OF SQUARES OF ERROR OF FIT =   8.1068E 02

.....0-N...........DENOM COEFFS.....
     0                          1.0000E 00
     1                          4.8551E 00

....0-N............NUMER COEFFS.....
     0                          2.5850E 01

......REAL POLE.......IMAG POLE.....
     -2.0597E-01         0.0000E 00

GAIN=       5.32443

STEADY STATE GAIN=       25.85043
```

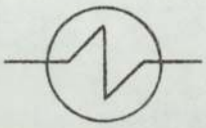
A5.4 SPECIMEN OUTPUT FROM PROGRAMME 'ESTOCS' (RUN 14)

```
RUN NUMBER: RW14
MAGNITUDE OF STEP INPUT: 29.98 DEG C

RESPONSE DATA FOR:
SHELL INLET
INITIAL TEMP= 20.65 DEG C
FINAL TEMP= 50.63 DEG C

GAIN=          .02298

      REAL ZERO            IMAG ZERO
     -3.9575E 02           0.0000E 00

      REAL POLE            IMAG POLE
     -2.9549E-01           0.0000E 00
      0.0000E 00           0.0000E 00

STEADY STATE GAIN=       29.98000

RESPONSE DATA FOR:
SHELL OUTLET
INITIAL TEMP= 20.54 DEG C
FINAL TEMP= 46.90 DEG C

GAIN=        -4.08359

      REAL ZERO            IMAG ZERO
      1.7403E 00           0.0000E 00

      REAL POLE            IMAG POLE
     -2.6960E-01           0.0000E 00
      0.0000E 00           0.0000E 00

STEADY STATE GAIN=       26.35999

RESPONSE DATA FOR:
TUBE OUTLET
INITIAL TEMP= 20.09 DEG C
FINAL TEMP= 25.30 DEG C

GAIN=         -.11442

      REAL ZERO            IMAG ZERO
      9.3783E 00           0.0000E 00

      REAL POLE            IMAG POLE
     -2.0597E-01           0.0000E 00
      0.0000E 00           0.0000E 00

STEADY STATE GAIN=        5.21000
```

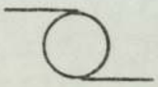
APPENDIX A6

AUTOMATING THE STEADY STATE TESTING OF HEAT EXCHANGERS - A PRELIMINARY ASSESSMENT

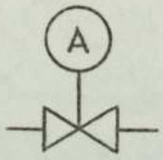
FIG. A6.1 GRAPHICAL SYMBOLS FOR FLUID CIRCUIT EQUIPMENT AND INSTRUMENTATION.



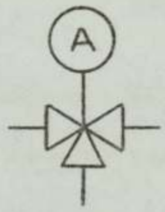
HEAT EXCHANGER



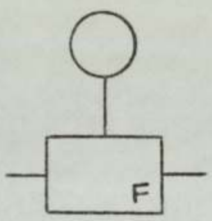
PUMP



VALVE & AUTOMATIC ACTUATOR
H ~ MANUAL VALVE



3 WAY VALVE & ACTUATOR



FLOW METER

STEAM TRAP

AUTOMATIC ACTUATING ELEMENT AND/OR POINT OF MEASUREMENT.
CODES :

FI FLOW INDICATOR
PI PRESSURE INDICATOR
TI TEMPERATURE INDICATOR
FIC FLOW INDICATOR & CONTROLLER
TIC TEMPERATURE INDICATOR & CONTROLLER

AUTOMATING THE STEADY STATE TESTING OF HEAT EXCHANGERS - A PRELIMINARY ASSESSMENT

A6.1. INTRODUCTION

This Appendix is intended to indicate the direction in which the company should proceed when it considers automation of the steady state testing of heat exchangers and to give the reader an appreciation of what this move would involve. The following discussion opens with the case for automated testing, then considers a number of alternative approaches to the problem and the hardware necessary to implement a solution. The implications of the introduction of an automated test facility into the company are presented and, finally, recommendations made.

The assessment was researched during the summers of 1978 and 1979.

A6.2. THE NEED FOR AUTOMATED TESTING

The company's Engineering Laboratory provides an essential service to the three operating divisions for development and approval testing of heat exchangers. All thermal performance tests are carried out on manually operated rigs which in most cases are not fitted with any type of automatic controller. Experience has shown that to obtain satisfactory test results it is necessary to establish a steady state test condition prior to and during the recording of test measurements. This normally results in a maximum of about 6 test points per working day being taken for a liquid to liquid heat exchanger. A typical test will require a minimum of 16 test points with additional repeat points being specified. Some tests may even require over 100 test points. When a heat exchanger is being tested, the test rig is in almost continuous operation for 8 hours per day and requires the full-time attention of a technician.

Until now the emphasis has been placed on reducing the time taken for the actual performance test to be carried out, i.e., the rig running time, while the time spent building the rig has not been considered. This latter part can take as long, or even longer, than the performance test itself. This is because often the basic test plant comprises solely of the pumps and heaters. To complete a test rig additional pipework is fitted to this basic configuration to suit the heat exchanger to be tested. Upon completion of the tests the pipework is stripped back to the basic plant because the pipework arrangement will usually need to be changed from one test to another and to make additional floor space available. However, the costs associated with building the rig are so much less than those incurred during actual testing that there is significantly more potential for cost savings in this latter area. Nevertheless, the time spent in rig building is an important factor because it can contribute significantly to the total testing time. To minimise the rig up time an automated rig could be designed so that it would only be necessary to change the input and output fittings to the test unit.

Clearly, thermal performance testing can form a large proportion of the laboratories work load, with the associated running costs being incurred. Consequently, there is a need to reduce testing time.

It is anticipated that a reduction in testing time would produce the following major benefits:

1. A quicker turn-round in testing resulting in -

(i) Performance data made available more rapidly.

(ii) Development time cut.

(iii) Urgent testing for customers could be performed with greater ease than at present.

(iv) Flexibility in laboratory work loading.

(v) The opportunity to make quality control checks on heat transfer performance practical.

2. Prospects of reduced testing overheads and energy consumption (for the same amount of testing)

In theory, automated testing has the potential to speed up testing by the application of control engineering principles. Further time savings can be expected by removing some of the uncertainties arising from human judgements made during manual tests and incorporating preset value judgement conditions to be satisfied in a controller. This does not mean that the technician's work is being replaced entirely by machine, it is merely shifting the emphasis in his work from the repetitive and perhaps uninteresting routine of testing to the preparation of instrumentation and then relieving him to carry out other work.

A6.3. GENERAL REQUIREMENTS FOR AN AUTOMATED TEST RIG AND SECONDARY BENEFITS

The prime requirements for an automated test rig would be to provide heat transfer and pressure loss data which is at least as accurate, reliable, and repeatable as data obtained from a manual test and give a significant improvement in testing time with the prospect of reduced costs.

Secondary benefits would include updating the laboratory facilities. This would significantly add to the company's existing technical and testing knowledge and help provide sound technological support for the future. Because of this, the company would be better placed to investigate and exploit new innovations. A well equipped test facility would have extra prestige value and demonstrate to customers the company's commitment towards improved products and the product backup which is available.

A6.4. OVERVIEW AND GENERAL ASPECTS OF THE PROBLEM

In order to develop a clearer mental picture of the problem it is useful to consider the case of an ideal automated test rig and then to obtain an appreciation for some of the control aspects.

A6.4.1. AN IDEAL AUTOMATED TEST RIG

An ideal automated test rig could be envisaged to be one which sets up all the desired test conditions, logs the measurements, analyses the data and outputs the results in the required form. In more detail, this would necessitate the input of the required test conditions - flow rates and temperatures - and the physical data for the test unit for the analysis of the results. The controller would then undertake the complete running of the tests. This would include the start up and shut down procedures, the control of the sequencing for changing flow rates, readjustment of flow rate and temperature set points, adjustment of the control parameters, performing the control functions according to the control algorithms and initiating data collection.

A6.4.2. SOME CONTROL ASPECTS

A problem area associated with the large flow turndown ratios (200:1 for water and 130:1 for oil, section A6.5.2) and also with the temperature range (2 - 125 ^{o}C) is the control aspects. The range of flows and temperatures which the rig must be capable of providing means that a wide variation in the control parameters which define the control and response of the system must be expected. Controller settings which provide satisfactory control under one set of test conditions can be expected to provide poorer control under different test conditions and under some conditions the control performance can deteriorate to such an extent that the system becomes unstable. How the control parameters vary and to what extent will be governed not only by flow rates and temperatures but by the dynamic behaviour of the equipment in the test rig - pipework, heaters, valves, instrumentation etc., - each have their own associated gains, capacities and transport lags which combine to give the dynamic behaviour of the whole rig. Each time a different heat exchanger to be tested is connected into the system the dynamic behaviour of the system can be expected to change. The system must also be able to cope with any disturbances which are introduced due to load changes external to the test rig, for example, disturbances in the steam supply due to load changes elsewhere in the laboratory and temperature changes in the water supply.

A6.5. PRACTICAL CONSIDERATIONS FOR AUTOMATED TESTING

A6.5.1. CONTENT OF A THERMAL PERFORMANCE TEST

The following procedure for an oil to water performance test is generally applicable for testing all two fluid heat exchangers.

In any typical performance test it is necessary to set a series of oil flow rates for each of a series of water flow rates, while maintaining constant either the desired inlet temperatures to the test unit or the mean bulk temperatures of the unit. When steady state conditions have been established at the desired temperatures and flow rates, the inlet and outlet temperatures at the test unit, and the flow rates are recorded. Pressure loss measurements across the oil and water sides of the test unit are also taken during the heat transfer tests. On completion of, or prior to the heat transfer tests, isothermal pressure loss tests are carried out on both the oil and water sides of the test unit. Both isothermal pressure loss tests are carried out for a series of flow rates at specified temperatures. Water isothermal tests are usually carried out at ambient temperatures with stationary oil on the other side of the test unit. For oil isothermal tests the water is usually drained from the unit.

A6.5.2. DUTY REQUIREMENT OF AN AUTOMATED TEST RIG

Before a technical specification can be drawn up detailing flow and temperature ranges etc., for an automated test rig, it is useful to examine the company's present thermal performance test facility and use this as an indicator to the type and the size range of heat exchangers which may be considered desirable and practical to test on a single rig.

There are 4 test rigs which together cover the entire flow range and heat inputs demanded for the testing of both water to water and oil to water shell and tube heat exchangers. The smallest unit might only be 50 mm diameter by 100 mm long while the largest maybe 1.35 m dia. x 4.3m in length. For applications utilising air there are 2 test rigs. The first, incorporates a plenum chamber for testing

APPROXIMATE SIZE OF HEAT EXCHANGER BORE DIA (mm) x TUBESTACK LENGTH (mm)	FLOW RANGE (l/min)		HEAT DISSIPATION RANGE (kW)	MAXIMUM PRESSURE LOSS ACROSS UNIT (kN/m^2)		MASS OF UNIT (kg)
	OIL	WATER		OIL	WATER	
1. 63 x 76 *	14-50	14-50	2.4-6	140 (includes 35 filter loss)	18	?
2. 105 x 305 *	45-136	68-159	20-43	with filter 520, without filter ≤ 170	45	?
3. 152 x 457 *	186-264	750-1318	89-122	110	22	?
4. 89 x 150 - 455 +	27-159	10-75	7.5-65	345	55	6.5-10.8
5. 127 x 231 - 767 +	23-259	23-182	15-180	380	35	12.7-25.5
6. 95 x 152 +	12-59	32-223	2.4-6.7	<240	<140	14
7. 117 x 279 +	15-75	59-423	7.7-23	<345	<140	25
8. 156 x 1499 +	79-395	98-682	60-185	<345	<140	100

* OIL INLET TEMPERATURE 125°C, WATER INLET TEMPERATURE 90°C

+ OIL INLET TEMPERATURE 80°C, WATER INLET TEMPERATURE 20°C

FLOW TURNDOWN RATIO ON TYPICAL HEAT EXCHANGER:
OIL - 10:1 MAX, NORMAL $\gtrsim$ 6:1
WATER - 10:1 MAX, NORMAL $\gtrsim$ 7:1

TABLE A6.1. TABLE GIVING AN INDICATION OF THE SIZE OF HEAT EXCHANGERS TESTED ON THE CURRENT 3 INCH PLANT & THE FLOW RANGES & HEAT INPUTS REQUIRED

	WATER CIRCUIT (TUBESIDE)	OIL (SHELL TX30) CIRCUIT (SHELLSIDE)
TEMPERATURE (^{o}C)	80 MAX	125 MAX
FLOW (l/min)	4.5 - 910	4.5 - 590
PRESSURE LOSS, PORT TO PORT, ON TEST UNIT (kN/m^{2})	170 MAX	345 MAX (PROVIDED FILTER NOT FITTED) USUALLY $<$ 170
HEAT DISSIPATION (kW)	200 MAX	

TABLE A6.2. PRELIMINARY SPECIFICATION FOR THE CAPABILITIES OF AN AUTOMATED TEST RIG radiator type units and can be connected to either an oil or a water circuit. The second, supplies high or low pressure air from 2 compressors for either air to air or air to water applications. There is one test rig enclosed in a cell used specifically for testing aero-engine heat exchangers.

How often and for what period of time each rig is in operation varies considerably. For example, the rig for testing the largest shell and tube units has not been used in the past 2 years, while the rig for heat exchangers at the other end of the size range is in regular use.

In the opinion of the Laboratory Manager the test rig which is in significantly more frequent use than any other rig, (order of 75% of all heat transfer tests over the past 3 years, carried out on it) is the oil/water plant for testing the small size shell and tube type heat exchangers (the 3 inch nominal pipe size plant). Therefore, it seems reasonable to use the capabilities of this rig as an initial guide. Table A6.1 indicates the range of heat heat exchanger sizes which have been tested on this plant and the flow ranges and heat inputs required.

Using the information in Table A6.1 and the flow and temperature ranges available on the existing 3 inch plant, a preliminary specification for the capabilities of an automated test rig can be specified. This is given in Table A6.2.

A6.5.3. GENERAL EQUIPMENT AND INSTRUMENTATION

This section discusses some of the major factors which will influence the selection of equipment and instrumentation for the oil and water circuits of the test rig.

A6.5.3.1. METHOD OF HEATING

It is likely that the choice of supplying heat to the two fluid circuits would be restricted to steam and electricity or a combination of the two.

Electric heaters may prove easier to control than steam heaters since both voltage and current can be accurately measured and regulated. However, disadvantages would be the long period of time required to raise the fluid temperature and the very high capital and running costs. It would also be necessary to install additional electrical equipment in the laboratory sub-station to meet the increased power demand.

Unlike electric heating, steam has an inherent safety feature; provided the steam is not superheated, the temperature to which the fluid can be heated cannot exceed the saturation temperature of the steam at the appropriate steam pressure. Recently, a new oil-fired steam boiler was installed for the laboratory making steam readily available and free from load disturbances caused by fluctuations in the demand for steam elsewhere within the company. Thus, the company can be expected to insist that this plant is used.

From the foregoing, it seems unlikely that heating by the combined use of steam and electricity would be attractive. Therefore it is assumed that the mode of heating will be steam.

The steam capacity required to provide the necessary heat input to the test rig is an important consideration. Firstly, there must be sufficient capacity such that disturbance effects introduced into the rig by the demands for steam elsewhere in the laboratory are minimised. It may prove necessary to ensure that should a demand for steam occur during the final period of rig stabilization or during the recording of test measurements, then the response is delayed until the test point is completed. Secondly, there should be sufficient capacity above the steady state requirement to allow the rig to be "forced" relatively quickly to the steady state condition. The speed at which this can be achieved will be limited by the size of the steam plant available, the cost of the control equipment and the onset of mixing problems caused by heating the fluid too rapidly. If the heat input is too high for a short period of time, a finite portion of fluid will be heated to a temperature above that of the remaining fluid in the circuit and this slug of hot fluid will travel round a closed circuit for a considerable time without mixing.

A6.5.3.2. HOW TO HEAT THE FLUIDS USING STEAM

In oil to water heat exchange applications, it is usually the oil which is being cooled. Therefore, during a performance test, the oil must be heated to the required temperature which, from a practical view point, would necessitate operating the oil on a closed circuit. Cooling water would normally be supplied from the laboratory's own pond or from a header tank and for testing could be used on either an open or a closed circuit. On open circuit, the heated water will be returned to the supply pond and wasted. When the larger rigs are running, the large heat dissipation rates can cause the supply water temperature to rise significantly and consequently acts as an input disturbance to the water circuit. Additionally, a heat input to the water circuit would be required to raise the water temperature to the desired value. Closed circuit operation would be more efficient because the heat gained from the oil could be utilised to heat the water, so eliminating the need for a separate heat input to the water circuit. However, a means of cooling the water to prevent excess heating would be required. This could be achieved by either a heat exchanger or by bleeding some of the hot water to waste and allowing colder feed water to make up the loss from a header tank. This bleed-feed method would be a simpler and cheaper solution for regulating the water temperature than that requiring a heat exchanger. It would be possible to speed up the process of heating the water by placing an additional heater in the circuit. However, this arrangement has the disadvantages of increased cost arising from the heater and the associated equipment and increasing the complexity of the system dynamics due to the interaction of the two fluid circuits at the test unit.

Therefore, from the above, an initial scheme for the test rig can be envisaged as comprising the oil and water circuits operating on a closed circuit, each with a static head provided by a header tank and the heat input supplied to the oil. The test unit would serve as a water heater, transferring heat from the oil to the water, with the water temperature regulated by a bleed-feed arrangement, fig. A6.2. (The control of the oil temperature is discussed in section A6.5.4.2).

A6.5.3.3. HEATERS

The large turndown ratios for the flows and the heat dissipations presents a problem when sizing the steam heaters because their selection will influence their response to control action and so contribute to the overall performance

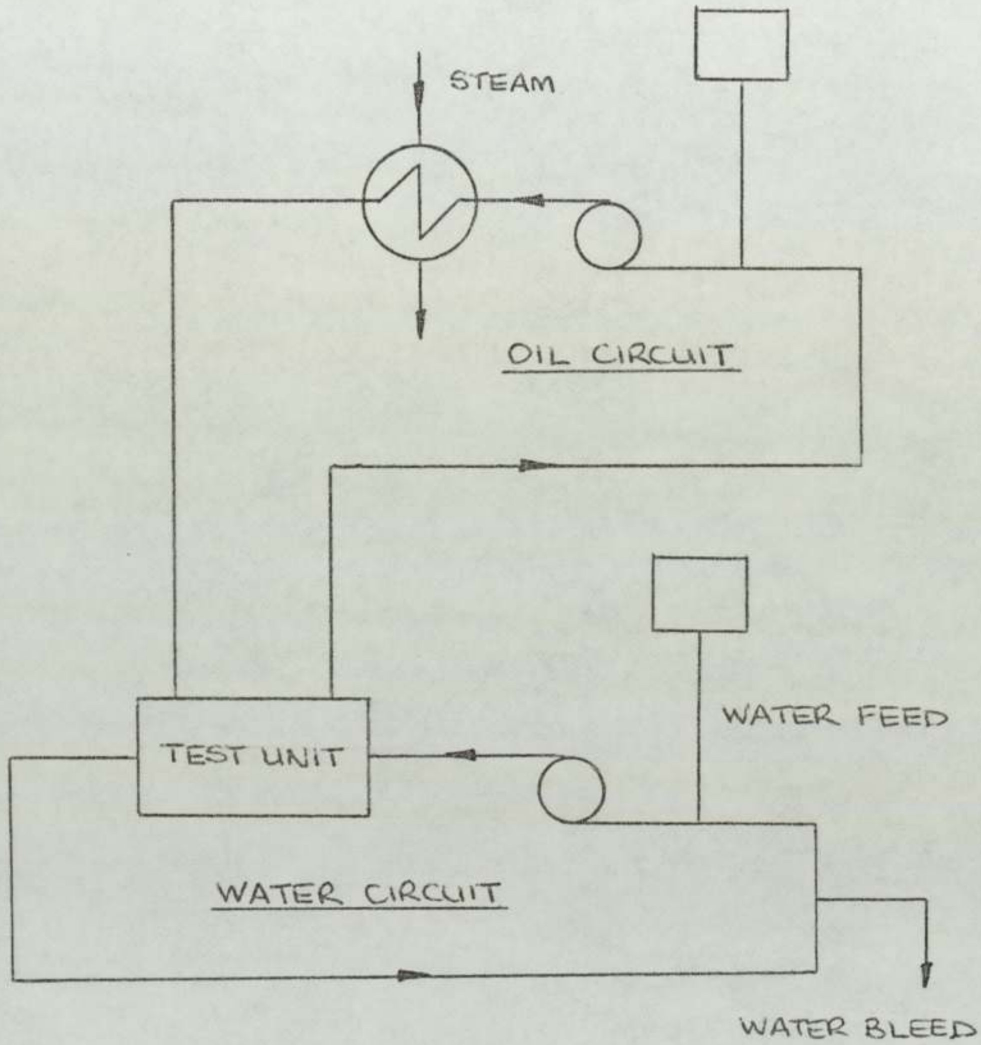


FIG. A6.2 SCHEMATIC FOR HEATING THE OIL AND WATER CIRCUITS - (TEST UNIT IN COUNTERFLOW OPERATION.)

of the test rig. A large steam heater may perform satisfactorily when high heat inputs are required but not when only low heat dissipations are needed. Therefore, it may prove an advantage to divide the heating and flow requirements between several heaters of different size. This aspect will need to be investigated further.

In situations whereby steam heaters are arranged to operate in parallel, a pump should be dedicated to each heater to prevent the possibility of instability (48).

Condensate removal and control can be a problem, particularly when a steam heater is operated over a wide load range. This is discussed in (34).

A6.5.3.4. CONTROL VALVES

Probably the most common final control element in processes today is the control valve. Wolter (54) says, "Proper control valve sizing is the single most important factor in the valves contribution to control". The important parameters to consider when selecting control valves are the minimum and maximum flow rates required and the available pressure drop, plus any special start-up or shut-down procedures. Where there are wide or rapid load changes, as can be expected in the automated test rig, it is best to retain higher pressure drops in order to assure satisfactory control. However, higher pressure losses mean a greater pumping power requirement and consequently the control valve pressure drop needs to be considered in relation to the pump and system flow versus head characteristic.

To accommodate the wide flow range specified for the test rig it is probable that several control valves of different size will be required.

Equal percentage type valves are usually specified for control because their gain characteristic compensates for changes in process gain and thus helps to ensure system stability.

A6.5.3.5. PUMPS

The wide flow variation creates problems for pump selection because it is desirable to maintain a reasonable overall efficiency for the pump over the entire flow range. For this reason it may be advantageous to use pumps of differing capacities arranged in parallel and which could operate alone or together.

The delivery head required of the pump(s) will be determined by the maximum flow rate and the associated fluid system losses - the summation of the pressure drops along pipes and across valves, fittings, steam heaters, and the test unit etc. As mentioned in the previous section, the pressure loss available for the control valve is an important factor which needs to be allowed for when selecting the pump(s). When pumps operate in parallel their characteristics need to be matched correctly to ensure satisfactory operation.

An obvious choice for water pumps is the centrifugal type. The flow can be controlled by throttling a valve downstream from the pump. However, it is wise to keep the pressure downstream of the valve above atmospheric and ensure that the minimum pressure in the valve is not sufficiently low to cause cavitation or air to come out of solution.

Oil can be pumped by both centrifugal and positive displacement type pumps. The flow from an oil centrifugal pump can be controlled in the same manner as described above. Extra precautions need to be taken when using a centrifugal pump to pump high viscosity oil in order to prevent overloading. Overloading could occur on start up on a cold day.

Flow from a positive displacement pump can be regulated using a by-pass valve or a variable speed motor, both with a throttle valve downstream to provide back pressure. Care should be taken when using a by-pass arrangement to prevent the pump overheating. Positive displacement pumps require protection equipment to prevent the sudden pressure build up which will occur should the flow stop.

These problems should be overcome by early discussion with pump suppliers.

A6.5.3.6. TEMPERATURE MEASUREMENT

The practical difficulties of temperature measurement, in particular the problem of ensuring good mixing of the fluid, have been discussed in I.3.2. In thermal performance tests it is the temperature at the test unit inlets and outlets or direct measurement of the appropriate temperature differences which need to be accurately measured. In the present steady state testing method the temperature measurement accuracy is assumed to be $\pm$ 0.05K (see Appendix A3.1). Therefore, the instruments and measuring equipment for the automated test rig must give an accuracy of at least $\pm$ 0.05K.

The following discussion is confined to high accuracy temperature measurement devices most suited to data acquisition. This narrows the field for practical purposes to resistance thermometers, thermistors, and thermocouples.

Resistance thermometers are capable of very high accuracy and depend upon measuring the change in resistance of a temperature sensitive resistor. It may be noted that the platinum resistance thermometer is used for the International Practical Temperature Scale of 1968 between the triple point of hydrogen and the freezing point of antimony.

Thermistors are semi-conductor devices with a negative temperature coefficient of resistance and a sensitivity many times greater than a resistance thermometer.

In general, for resistance thermometers and thermistors, the measuring equipment required for high precision work is expensive. The accuracy of these two devices with the appropriate measuring instruments and calibration can be better than $\pm$ 0.01K and will, therefore, meet the accuracy demanded for temperature measurement for an automated test rig. They also have the advantage of giving a direct measure of absolute temperature and have better long term stability than thermocouples.

In principle, thermocouples provide a simple method of temperature measurement and rely on the thermoelectric effects produced when two dissimilar metals are joined at the ends and the junctions are subjected to different temperatures. It is usual to maintain one junction of the thermocouple at a reference temperature and for convenience a mixture of ice and water at $0^{o}C$ is often chosen. The practical problems of using thermocouples for precision work are not to be under-estimated; see references (2) and (5). Unlike resistance thermometers and thermistors, thermocouples only give a relative indication of absolute temperature; however, this can be used to advantage, for example, to measure temperature difference directly, although for accurate work calibration may be a problem.

All three devices described above have a good transient response but this is affected by the size of the measuring element and the shielding around it.

Before a particular method of temperature measurement can be chosen for the automated test rig it will be necessary to investigate fully the three methods described above, with particular reference to installation difficulties, operating precautions, sensor and measuring cost, and calibration requirements.

Additional reading can be found in (14) and (38).

A6.5.3.7. FLOW MEASUREMENT

The accurate measurement of flow rate is essential in heat transfer performance tests. This section considers the main types of flowmeter which can be used to measure the flows required on the automated test rig and suggests those which could be most suitable.

ORIFICE PLATE AND VENTURI TUBE

The main disadvantage of both the orifice plate and venturi tube is their limited operating flow range; only 3 or 4:1. This is because of the square-root relationship between flow rate and differential pressure. A further disadvantage of the orifice plate is the high irrecoverable pressure loss. The venturi tube has only a fraction of the orifice plate's pressure loss but is expensive in comparison. The calibration of orifice plates for water flow can be predicted to about $\pm$ 1% from the appropriate British Standard, or they can be individually calibrated to give greater accuracy. Under most operating conditions the devices maintain their calibration over long periods of time. However, significant errors can occur if they become damaged.

Both meters can be used to measure the flow of water and oil.

TURBINE METERS

The flow range of a turbine meter is typically 10 or 15:1, although for specialist applications one manufacturer quotes an extended range for liquids up to 75:1. Linearity over the operating range is good, generally better than $\pm$ 0.5% of point. Turbine meters are capable of giving very high accuracy provided they are calibrated at frequent intervals and have a short term repeatability better than $\pm$ 0.1% of point. The performance of a turbine meter is very sensitive to viscosity changes, upstream flow disturbances, bearing wear and meter damage resulting from "dirty" fluids. The head loss is between 1.5 and 2 velocity heads.

VORTEX METERS

The vortex meter is based on the natural phenomenon known as vortex shedding. The frequency with which vortices are shed from alternate sides of a bluff body is proportional to the fluid flow rate.

The minimum acceptable flow rate for the vortex meter is usually governed by the fluids Reynolds Number which should be $>$ 10000. The flow range is typically 15:1, although the maximum flow rate is limited by the onset of cavitation. Repeatibility is $\pm$ 0.15% of point. As there are no moving parts, the vortex meter retains its calibration and because the phenomenon of vortex shedding depends only on the fluid velocity and not the fluid properties it is not necessary to calibrate under actual operating conditions or with the actual fluid. However, the effects of viscous fluids on performance have yet to be fully determined. Pressure loss is about 2 velocity heads.

POSITIVE DISPLACEMENT METERS

The flow turndown of positive displacement meters can be up to 100:1. Their accuracy is practically unaffected by upstream flow disturbances. Calibration is maintained over long periods of time, with repeatability better than $\pm$ 0.25%. The main disadvantages are their high head loss and for the same flow range they are larger and more expensive than the turbine meter.

ELECTROMAGNETIC FLOWMETERS

Electromagnetic flowmeters work on the principle of electromagnetic induction and therefore are only suitable for fluids having reasonably high electrical conductance. For example, water. They are unsuitable for oils. The flow turndown is typically 10:1 and calibration accuracy can be better than $\pm$ 1% of reading. However, the calibration does tend to drift with time and for this reason the meter is generally regarded as being unsuitable for flows $<$0.5 m/s. A further disadvantage is that they are expensive, particularly for small pipe sizes. Pressure loss is very low because the meter imposes no obstruction to flow.

ULTRASONIC FLOWMETERS

Ultrasonic flowmeters deduce the fluid flowrate from the effect the flowing fluid has on the velocity of sound in the fluid. The time difference between two successive ultrasonic pulses in either direction across a pipe is measured and is proportional to the flow rate. Accuracies of the order of $\pm$ 1% of reading may be expected for not too severely distorted velocity profiles. This accuracy figure places a restriction on the minimum flow rate which can be measured. The maximum flowrate is limited only by practical considerations. However, too wide a low range can reduce the accuracy. The calibration is maintained for long periods of time and the repeatability is quite high. The ultrasonic flowmeters chief disadvantage is the very high cost of the associated equipment, especially for small pipe sizes. The meter can be used to measure both water and oil flow and since there is no flow obstruction, head loss is negligible.

INSTALLATION OF FLOWMETERS

Care must be taken in installing flowmeters because large measurement errors can be caused by swirl producing devices - pumps and bends - at inlet to the meter and the presence of bubbles in the liquid as it passes through the meter. Orifice plates, turbine meters, and vortex meters are all sensitive to swirl at their inlets and therefore it is essential to ensure that a long length of straight pipe of the same diameter as the meter inlet is placed immediately upstream of the inlet to "calm" the flow. Some manufacturers of these flowmeters recommend 10-15 pipe diameters of upstream straight pipe. However, in some cases, even 50 diameters may not be sufficient. Where it is not practical to install a very long calming section, a flow straightener may be used, but in this case it is essential that at least 10 diameters of straight pipe is placed between the flow straightener and the meter.

Errors in flow measurement will result if bubbles form in the liquid and are present as the fluid passes through the meter. Bubbles can be formed in three ways; by mechanical entrainment, cavitation, and the release of dissolved air from solution. Mechanical entrainment can be caused by the pump sucking air into the fluid and, in recirculating systems when bubbles are allowed to be returned to the system by the pump. Cavitation occurs when the pressure of the liquid falls below the vapour pressure of the liquid and local boiling occurs. The release of dissolved air is usually only a problem in systems using oil and may occur at any pressure below 1 atmosphere absolute. It is interesting to note that oil under normal room conditions usually contains about 8 or 9% by volume of dissolved air. Cavitation and the release of dissolved air can often be prevented by the use of a back pressure valve to ensure that all points in the system are kept above atmospheric pressure, however, it should be remembered that the act of closing the valve may lower the pressure in the valve sufficiently for bubbles to form.

The most suitable location for the flowmeters is before the inlet to the test unit where the fluid pressure is reasonably high and the fluid temperature is usually fairly constant.

FLOWMETERS FOR THE AUTOMATED TEST RIG

On the basis of the above discussion, the turbine meter and the vortex meter appear to be the most suitable meter types to cover the wide flow range and give the high accuracy required.

For measuring the water flows, a single vortex meter plus one turbine meter may prove sufficient. The choice of a vortex meter eases the calibration requirements, while the turbine meter is required for the very low flow rates because a suitable vortex meter is not available. Oil flow measurement will probably require 2 or 3 turbine meters.

Correct calibration of flowmeters is essential and for an excellent discussion of this subject the reader is referred to reference (21).

Other useful reading can be found in (11), (20) and (32).

A6.5.4. CONTROL METHODS AND SCHEMES

The object of this section is to introduce the basic types of controller before discussing the application of them and more advanced control techniques to the problem of controlling the process variables on the automated test rig.

All practical control systems involve the feedback of information from the controlled variables for comparison with their desired states and for subsequent control. Fig. A6.3 shows in block diagram form the essential features of a control system; the arrow headed lines can represent any number of inputs or outputs.

The automated test rig is a multivariable system with interaction between the oil and water circuit control loops. This is a result of the heat transfer taking place between the two fluid circuits in the test unit. The complexity of the control problem can be simplified considerably by individually controlling as many of the input variables as possible. However, interaction can lead to oscillation in the controlled variables in both control loops. Whether or not it will be necessary to take into consideration the effects of interaction when developing the control strategy for the automated test rig will need to be determined.

A6.5.4.1. BASIC CONTROLLERS

The three prime modes of control action are proportional, integral, and derivative control.

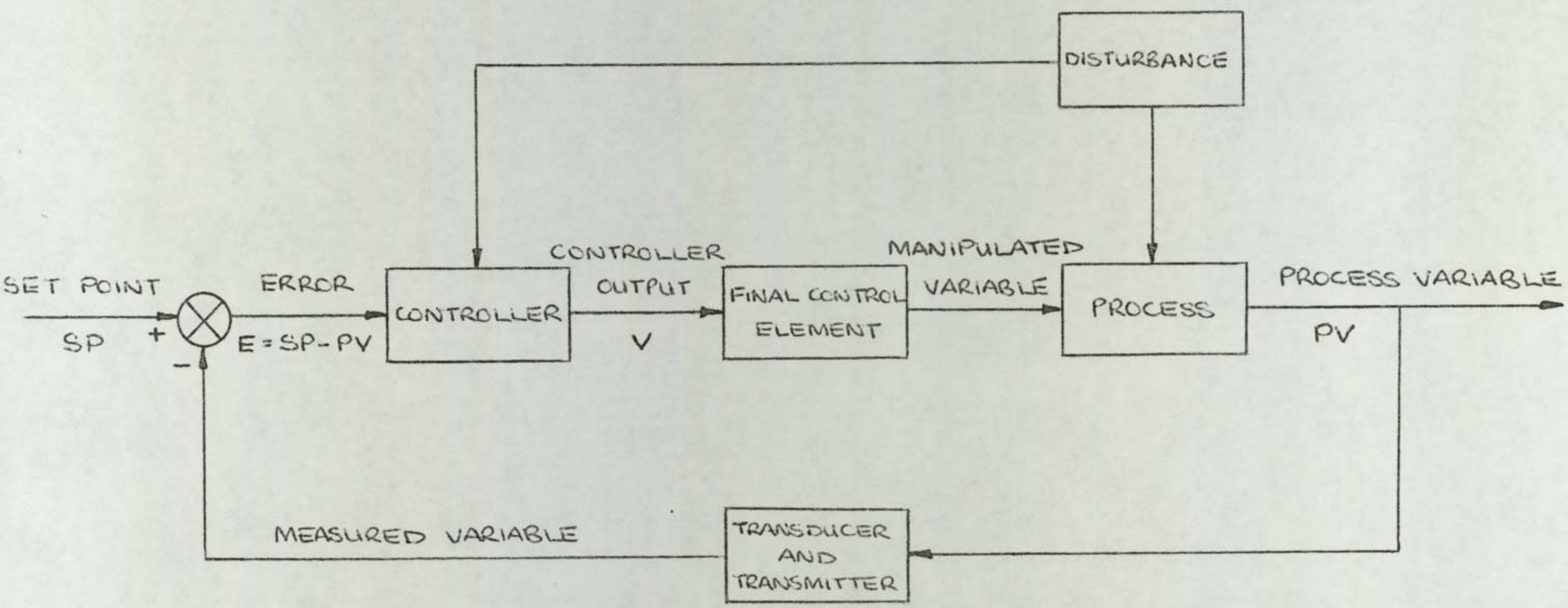


FIG. A6.3 BLOCK DIAGRAM OF A FEEDBACK CONTROL SYSTEM.

For a proportional controller, the output signal is proportional to the error signal. The equation expressing proportional control is:

$$V = K(E) + M \tag{A6.1}$$

where V = output signal to the final control element

$K = \frac{100}{PB}$ = proportional gain

E = SP-PV = error signal

SP = set point

PV = measurement of the process variable to be controlled

M = output signal from the controller when the error is zero; often called the "manual reset" term

PB = proportional band and is the percentage change in error required to move the final control element over its full range

Alone, proportional control is unsatisfactory if frequent load changes or disturbances occur in the process. It is characterised by a high maximum deviation and a significant time of oscillation of the controlled process variable and by offset. Offset, sometimes called steady state error, is the condition which exists after the transient effects of a load change or disturbance have disappeared and is the difference between the set point value and the actual value of the process variable. Offset can be reduced by increasing the gain, however, the extent to which this can be done is limited by the onset of instability in the control loop. It can be prevented by adjusting the manual reset term to allow the process variable to equal the set point under the new load conditions.

Integral (or reset) control action gives an output which is proportional to the time integral of the error and can be expressed as:

$$V = \frac{K}{T_R} \int_0^t E \, dt \qquad \text{(A6.2)}$$

where T_R = integral time or reset time.

When integral action is used in conjunction with proportional control, porportional plus integral control (P + I), the response of the controlled variable exhibits a higher maximum deviation, a longer response time, and a longer period of oscillation than with proportional control alone. Additionally, the integral term eliminates offset. P + I control is a frequently used combination, especially where the responses of other components in the control loop are rapid.

Derivative action (or rate control) gives an output which is proportional to the derivative (or rate of change) of the error and is given by:

$$V = KT_D \frac{dE}{dt} \qquad \text{(A6.3)}$$

where T_D = derivative time or rate time.

Derivative control is an anticipatory action and is employed where excessive oscillations have to be eliminated. Proportional plus derivative control (P + D) produces the least oscillation and the lowest maximum deviation of the controlled variable, but the same offset occurs as with proportional control alone with the same gain. However, the addition of derivative action allows a higher gain to be used before the control system becomes unstable and therefore a smaller offset can be obtained.

The controller often used because of its versatility, is the 3 term controller which combines the proportional, integral, and derivative modes, (P + I + D). It is a compromise between the advantage and disadvantages of P + I and P + D control. The mathematical relationship for a 3 term controller is the addition of equations A6.1 - A6.3:

$$V = K \left[E + \frac{1}{T_R} \int_0^t Edt + T_D \frac{dE}{dt} \right] + M \qquad (A6.4)$$

Fig. A6.4 compares the response of the controlled variable using different control modes following a step load disturbance.

A6.5.4.2. APPLICATION OF FEEDBACK CONTROL TO TEMPERATURE CONTROL OF A STEAM HEATER

Three term controllers are usually used for temperature control of heat exchangers. Derivative action is needed for long time lag systems and response to sudden load changes or disturbances and integral action is required to eliminate offset.

Perhaps the simplist form of control is shown in Fig. A6.5. Here the temperature of the fluid to be controlled is fed back to a controller for comparison with the set point and a signal sent to the control valve in the steam line to alter the valve opening, thus, changing the steam flow rate. The serious disadvantage of this method of control is that the effects of disturbances in the steam supply, e.g., fluctuations in flow and pressure, will be propagated through the steam heater before a change in temperature is sensed and corrective action is taken. This problem can be overcome by the application of cascade control, fig. A6.6. In this arrangement, the output from the temperature controller (the primary or master controller) adjusts the set point of the flow controller (the secondary or slave controller),

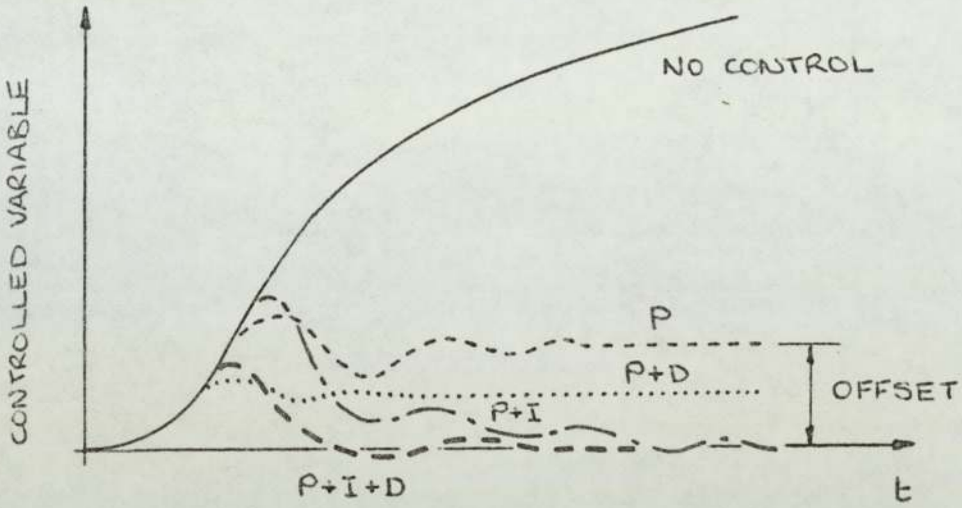


FIG. A6.4 COMPARISON OF THE RESPONSE OF THE CONTROLLED VARIABLE USING DIFFERENT CONTROL MODES FOLLOWING A STEP LOAD DISTURBANCE.

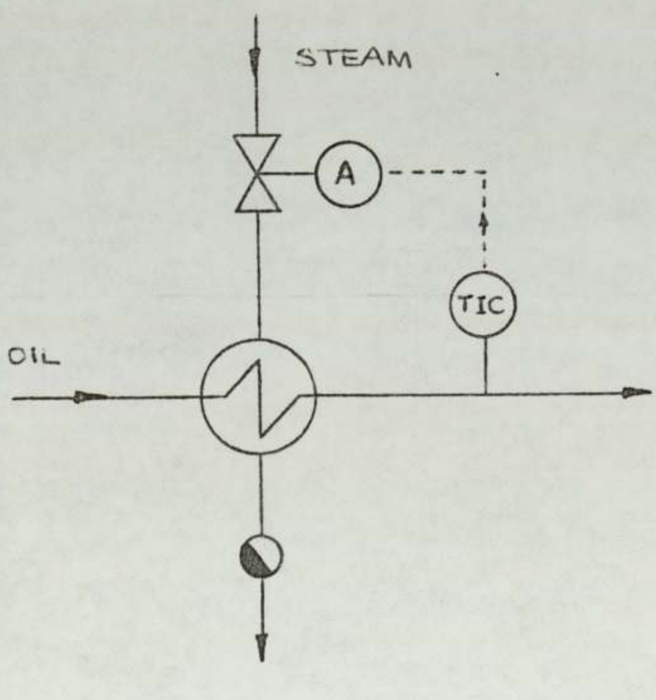


FIG. A6.5 SIMPLE FEEDBACK TEMPERATURE CONTROL OF A HEAT EXCHANGER.

thus, provided the secondary controller responds quickly to steam flow disturbances, these disturbances will be damped out and the controlled temperature will be unaffected.

When the response of the heat exchanger is too slow to hold the temperature deviation within desired limits, i.e., when temperature control is critical, the transient characteristics of the heat exchanger can be circumvented through by-passing part of the fluid stream around the heat exchanger and mixing the warmer fluid with the cooler fluid; fig. A6.7. This arrangement may cause problems which result from the time delay introduced because of the length of mixing pipe needed to blend the two fluid streams and the difficulty of ensuring complete mixing.

The disadvantage of feedback control on its own is that when a disturbance enters the fluid whose temperature it is desired to control, corrective action is not initiated until the disturbance has caused the controlled variable to deviate and corrective action is not felt until the changes produced by control action have propagated through the system. Therefore, this control technique alone maybe unsatisfactory for processes which have frequent disturbances of large magnitude, for example, set point changes in flow and temperature, and/or where large time lags exist which delay corrective action taking place. This disadvantage can be overcome by combining the advanced control technique of feedforward control with feedback control.

Feedforward control compensates for disturbances which would have otherwise caused a deviation in the controlled variable, later in time, by predicting the value of the inputs which are manipulated in order to keep the controlled variable at the desired value.

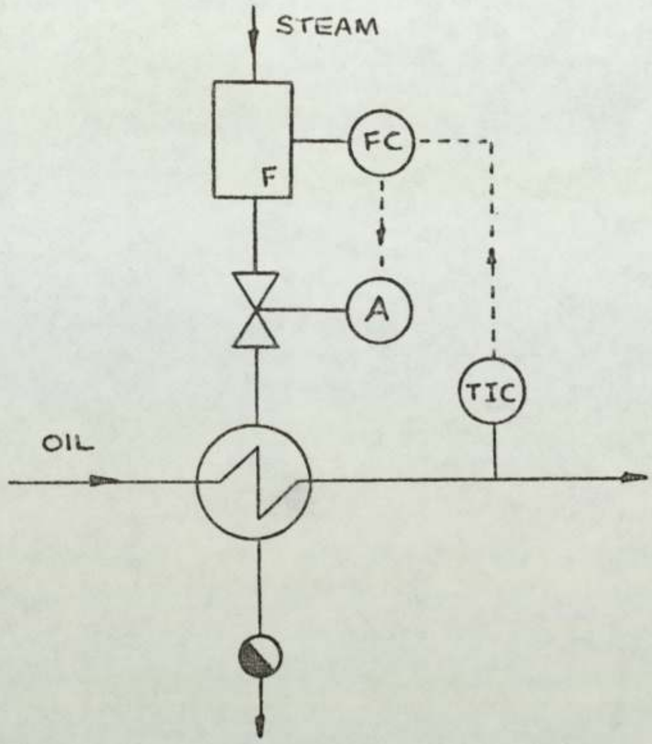


FIG. A6.6 CASCADE CONTROL OF OUTLET TEMPERATURE FROM A HEAT EXCHANGER.

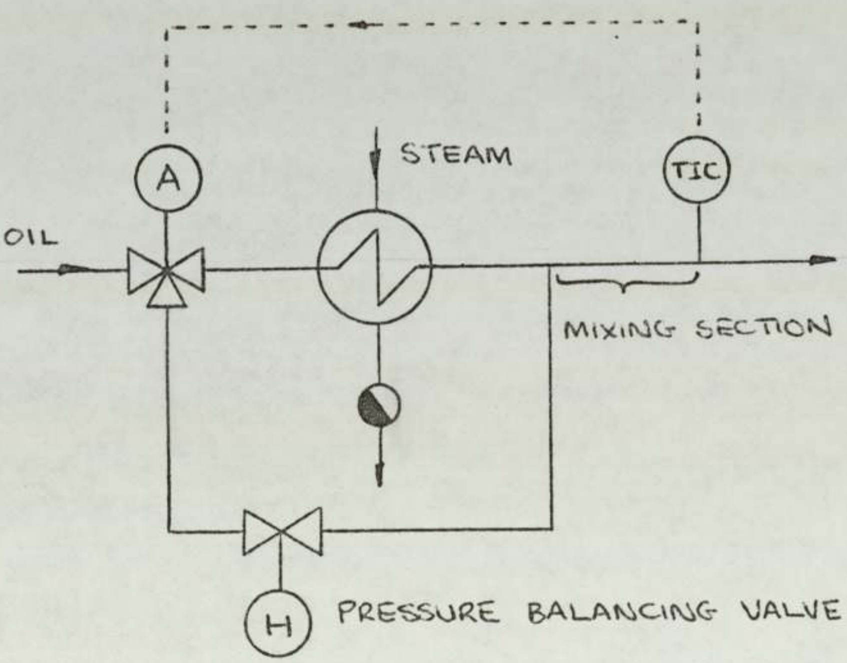


FIG. A6.7 BY-PASS TEMPERATURE CONTROL OF A HEAT EXCHANGER.

A feedforward algorithm in the form of a simple steady state energy balance can give a significant improvement in heat exchanger control. A steady state heat balance for a steam heater gives the equation:

$$\dot{m}_f c_f\ (\theta_o - \theta_i) = \dot{m}_s \Delta h_s \qquad \text{(A6.5)}$$

where $\dot{m}_f$ = mass flow rate of the shellside fluid, kg/s

c_f = specific heat of the shellside fluid, kJ/kg K

θ_o = desired shellside outlet fluid temperature, ^oC

θ_i = shellside fluid inlet temperature, ^oC

$\dot{m}_s$ = mass flow rate of steam (on tubeside), kg/s

Δh_s = change in specific enthalpy of steam across heat exchanger, kJ/kg K

This equation can be solved for the steam flow, $\dot{m}_s$

$$\dot{m}_s = \frac{\dot{m}_f c_f\ (\theta_o - \theta_i)}{\Delta h_s} \qquad \text{(A6.6)}$$

which will give the desired shellside fluid outlet temperature.

The addition of feedback allows for deficiencies in the feedforward algorithm, for example, heat losses and slight errors in the fluid specific heat and the steam enthalpy change, so that the desired outlet temperature is actually reached. Thus, θ_o in equations A6.5 and A6.6 can be replaced by θ_o', where θ_o', the output from the outlet temperature controller is the temperature set point for the feedforward controller. The feedforward control scheme is shown in fig. A6.8.

The lead/lag unit included in the feedforward control scheme prior to the steam flow controller, is a dynamic

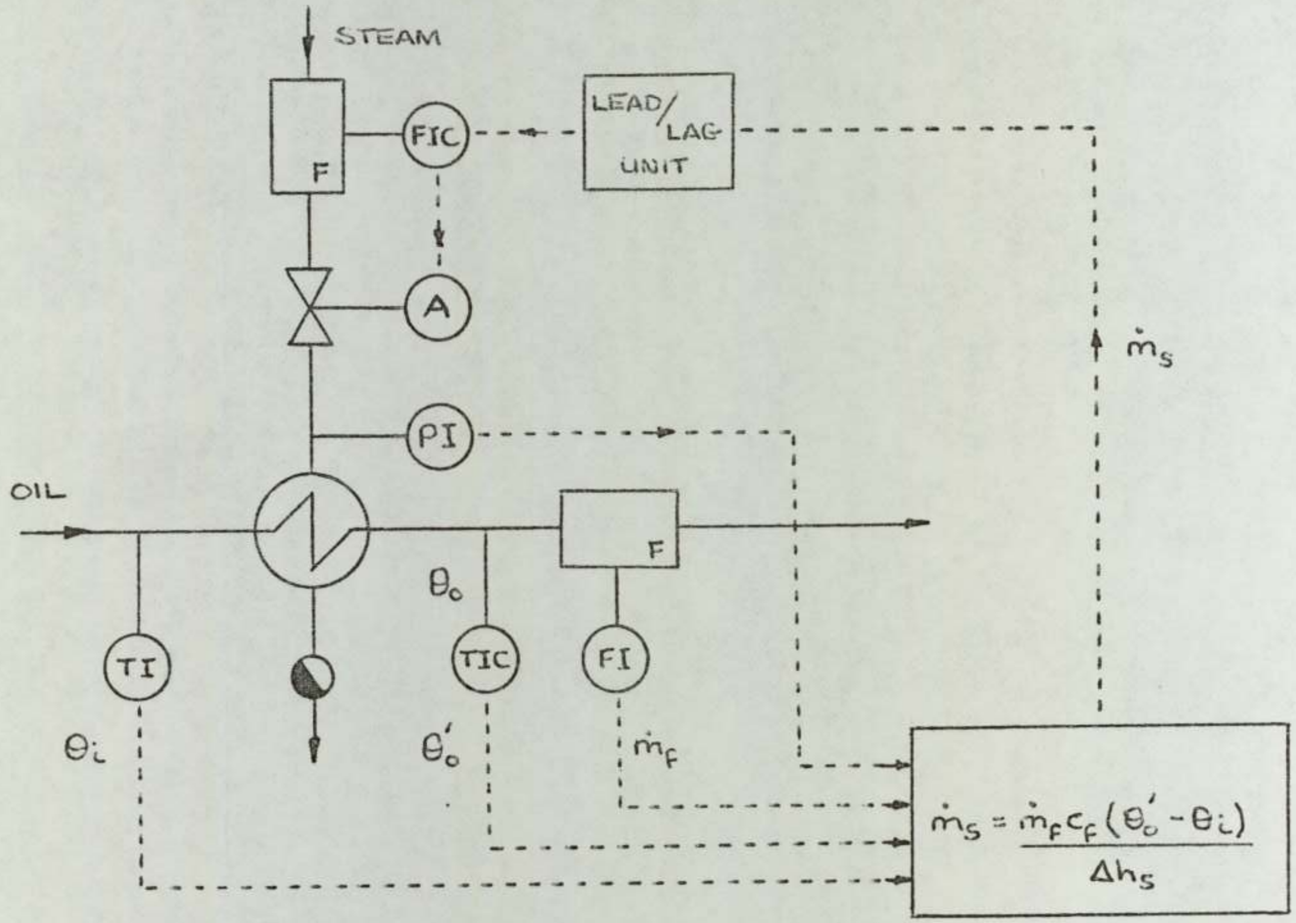


FIG. A6.8 FEEDFORWARD CONTROL OF A HEAT EXCHANGER.

compensation element. Its purpose is to reduce the overshoot and undershoot which can occur before the fluid outlet temperature settles at the desired value due to the difference in dynamic behaviour of the shell and tubeside fluids. Often, the lead/lag unit is of the form:

$$G(s) = \frac{T_2 s + 1}{T_1 s + 1} \tag{A6.7}$$

where $G(s)$ = transfer function of the lead/lag unit

s = Laplace operator

T_1, T_2 = adjustable time constants

In some cases the addition of a time delay element will improve the compensation.

Fig. A6.9 shows feed-forward control applied to the water circuit.

The disadvantage of feedforward control is that it is costly to implement and feedforward loops are more difficult to tune and operate than conventional feedback controllers.

An informative discussion on the control of heat exchangers is given in (30), while (52) compares seven different direct digital control algorithms for a heat exchanger.

A6.5.4.3. CONTROLLER TUNING TECHNIQUES

For all control systems it is necessary to determine suitable controller parameters which will define how the process will respond to control action, for example, values for gain, integral, and derivative times. This is often achieved by testing the plant, although initial estimates can be obtained by digital or analogue simulation of the process. In both cases either empirical or theoretically based tuning techniques are applied.

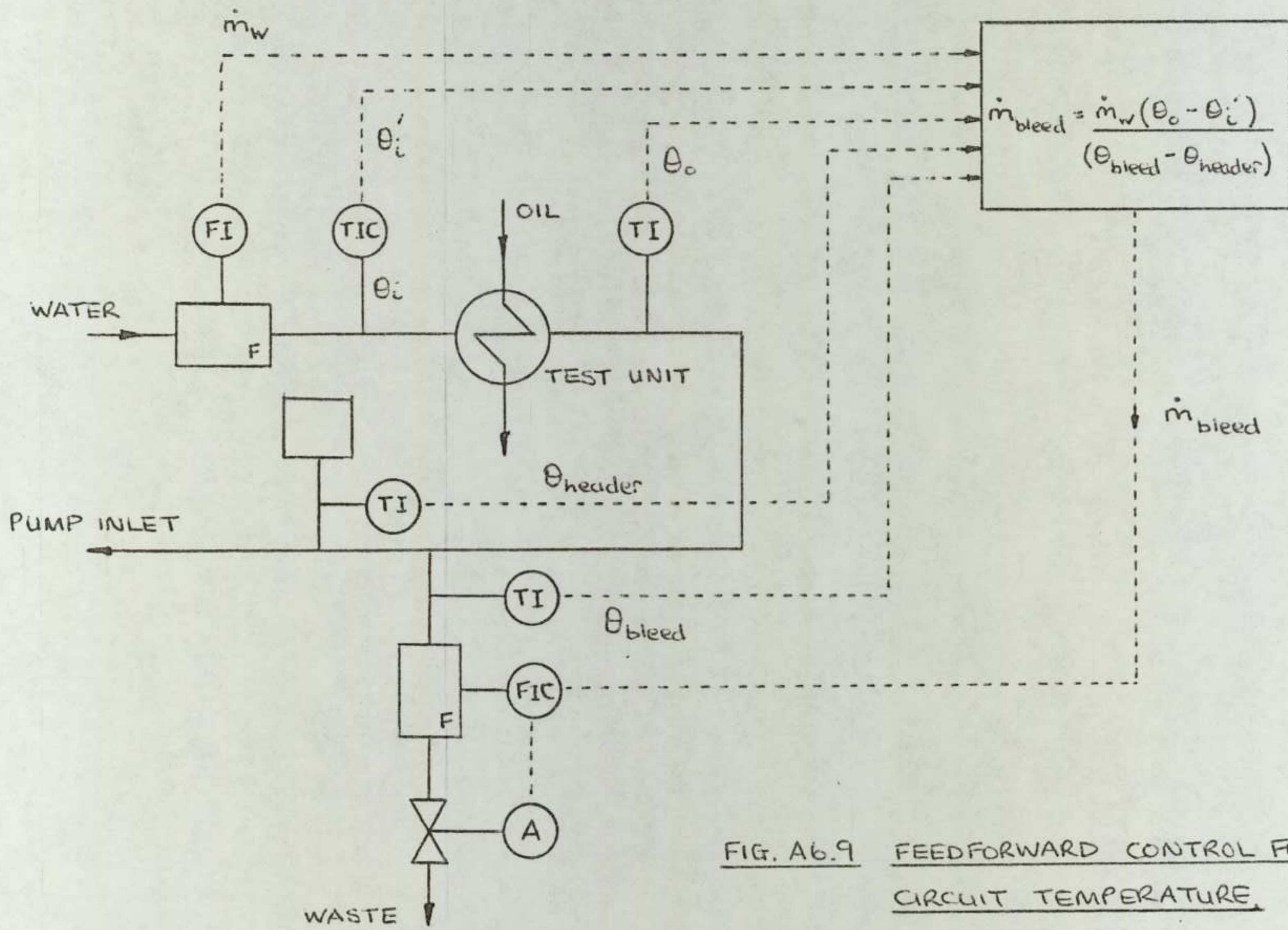


FIG. A6.9 FEEDFORWARD CONTROL FOR THE WATER CIRCUIT TEMPERATURE.

In the process industries the tuning criterion generally accepted as nearly optimum is "$\frac{1}{4}$ amplitude damping". The ratio of the overshoot of the first peak to the overshoot of the second peak one cycle later is 4:1 and is a compromise between a fast rise time and a reasonable settling time.

The empirical tuning method due to Ziegler and Nichols (56) is based on $\frac{1}{4}$ amplitude damping. The method first requires setting the integral time of a 3 term controller to infinity and the derivative time to a minimum, then adjusting the gain and introducing a step change. This is repeated until the process variable cycles with constant amplitude. From the corresponding gain and period of oscillation the controller settings are easily calculated from the relationships recommended.

A second technique is that of Cohen and Coon (10). The controller is set to manual and the open loop response recorded for a step input to the process. The response curve, called the process reaction curve, is generally "S" shaped (sigmoidal) and it is assumed that it can be approximated by a time delay followed by an exponential with a single time constant. The apparent time delay and the apparent time constant are determined from the response curve, fig. A6.10, and, together with the steady state gain of the system, used to calculate the controller settings for the desired control action from mathematically simple equations derived by the authors. Experience has shown that the controller settings obtained in this way give a fairly good closed loop response.

The controller settings obtained by these two methods should be treated as first estimates only. Finer tuning, by

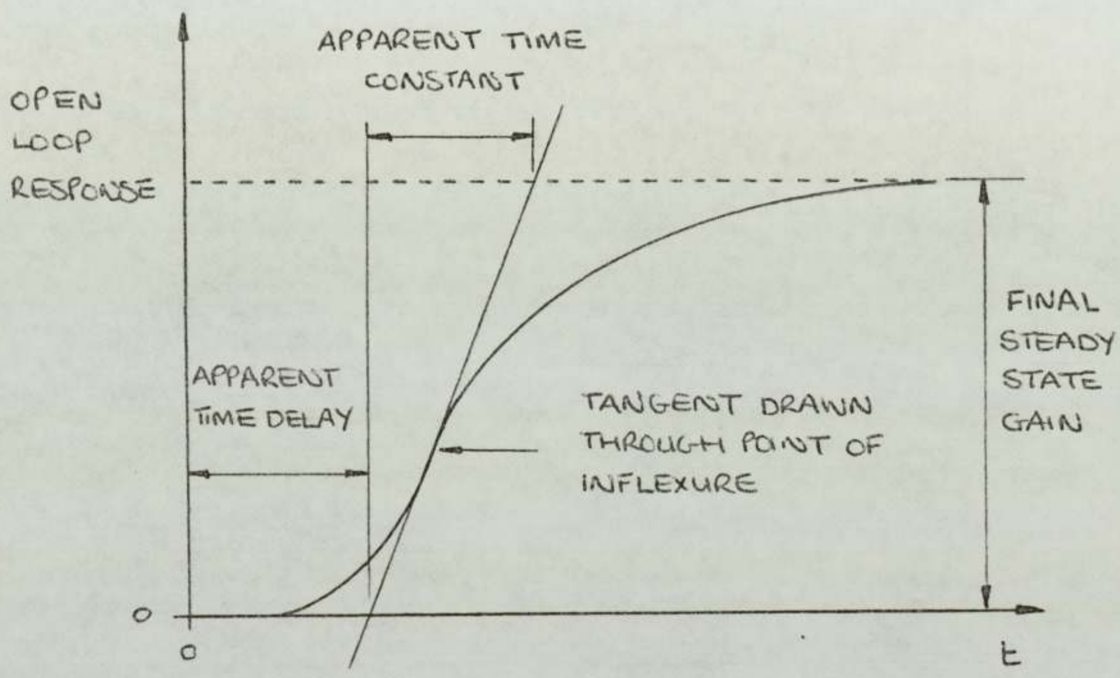


FIG. A6.10 PROCESS REACTION CURVE - (COHEN & COON TUNING METHOD.)

trial and error, will probably be necessary to achieve the "best" response for the particular process.

In theory, tuning techniques could be carried out online. However, the added complexity and difficulties involved may not justify this step being taken.

The above techniques are usually applied to continuous (analogue) controllers. Digital controllers are more difficult to tune because the sampling time must be taken into consideration. The control parameters for digital controllers can be optimised using an integral criterion based upon minimisation of the total error under the response curve. In general:

$$I = \int_0^\infty f\left[t, e(t)\right] dt$$

where I = integral criterion

t = time

e(t) = variation in error with time

Tuning graphs and tables based on integral criterion have been published. For a fuller discussion see (49).

It is important to note that optimal controller settings for load changes are not optimal for set point changes and vice versa, (see ref. 49, p177).

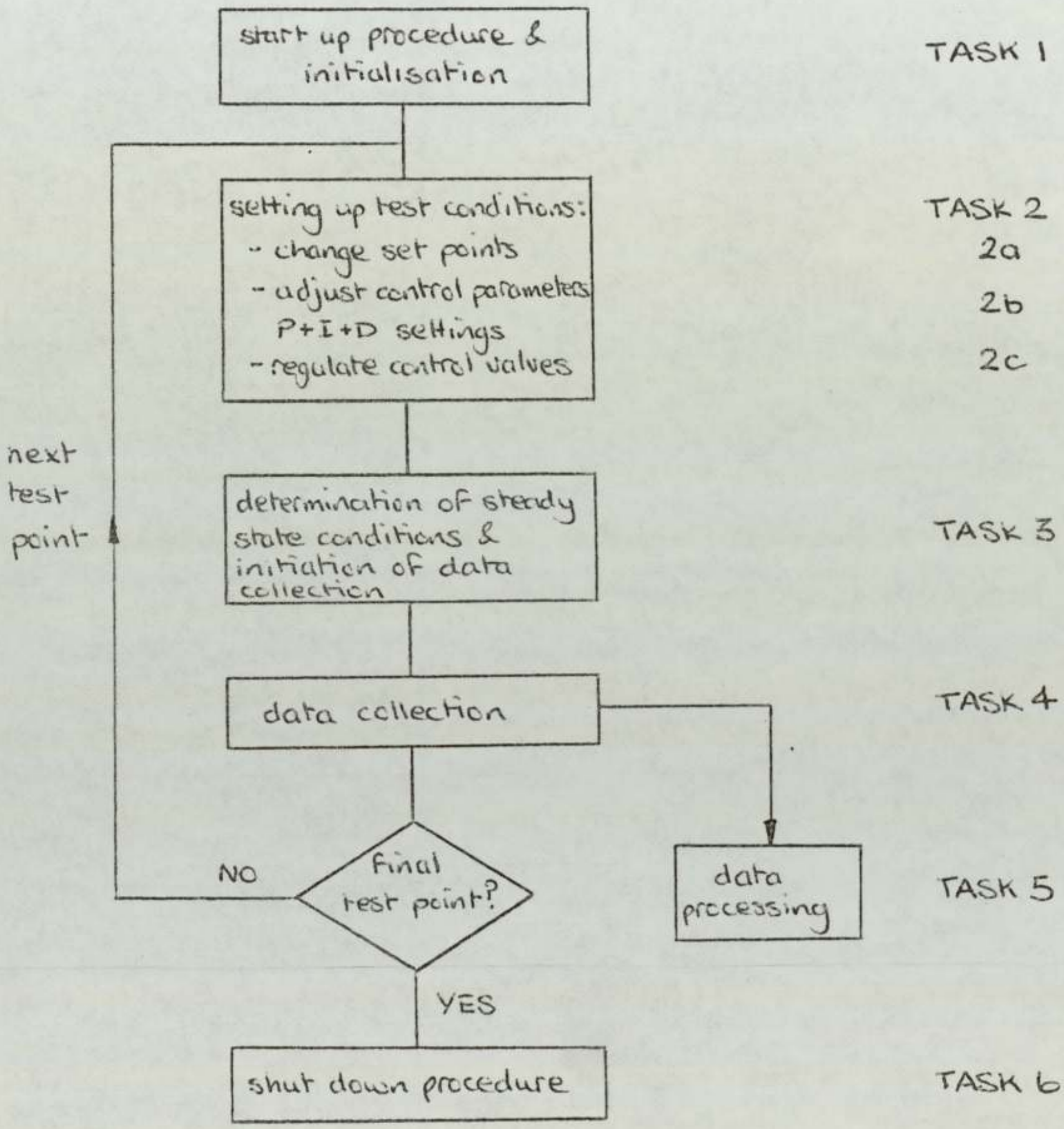
A6.6. THE DEGREE OF AUTOMATION

The two extremes in approach to automated testing can be considered on the one hand as total manual control (the present situation), and, on the other, total computer automated control. In the latter case, the operator would only be required to set up the instrumentation, input the data necessary for the test to the computer and press the start button; the computer would then supervise the whole test. Between these two extremes are numerous schemes for partial automation. However, they all have one important factor in common; the need for a human operator to be present for periods of time during the test to supervise the test and perform certain tasks.

Fig. A6.11 shows the operation of an automated test rig divided into 6 tasks. One possible scheme for a part automated test rig might be one incorporating 3 term, cascade, and by-pass controllers for flow and temperature control, with the operator manually changing the set points and control parameters, (tasks 2a and 2b). Task 2c would be done automatically by the controllers. The operator would also start up and shut down the rig, (tasks 1 and 6). Data collection (task 4) could be via a data logger outputting onto paper tape or magnetic tape for subsequent offline processing by computer (task 5). The determination of whether or not steady state conditions had been reached (task 3), could be carried out visually by the operator referring to a record of the test unit inlet and outlet temperatures, or by an electronic processing unit linked to the data collection device to monitor these temperatures. Data collection could be initiated by the operator or automatically by the processing unit. When data collection has been completed, the operator would be alerted to change the set points etc.

Because a wide variation in control parameters can be expected, part automation enables first-hand experience to be gained from tuning the controllers and gradually establishing a history of controller settings for different sized test units operating under differing flows and temperatures. This empirically derived knowledge will be valuable data when considering the degree of sophistication

FIG. A6.11 TASKS TO BE PERFORMED FOR THE RUNNING OF THE AUTOMATED TEST RIG.



necessary for a fully automated test rig and how far along the path to full automation the company might wish to proceed. If feedforward control is indicated, this necessitates the use of computing facilities to implement the control strategy. The use of the computer could be extended to include changing the set points and control parameters, the determination of steady state conditions, data collection and online processing. The inclusion of tasks 1-6 in fig. A6.11 under computer control leads to total computer automated control.

Digital computer controllers have several advantages over analogue controllers. Analogue controllers are inflexible because the control strategy is restricted to the one which can be implemented satisfactorily in analogue hardware and subsequent changes to the control strategy require modification of this hardware. Logical decisions are much easier to implement digitally than via an analogue system and the control strategy can be easily changed by changing the computer programme.

There are two control philosophies on which computer control of an automated system can be based; direct digital control (DDC) and a combination of digital and analogue control. They differ in that in DDC the computer carries out the control strategy and positions the final control element, whereas in the hybrid system the digital computer provides set points and control parameters for analogue control loops. DDC is costly and difficult to justify economically, and, in addition, it is usual to have analogue backup stations on critical loops. Both DDC and hybrid control can be extended to provide supervisory control for several test rigs or a number of different processes. Fig. A6.12

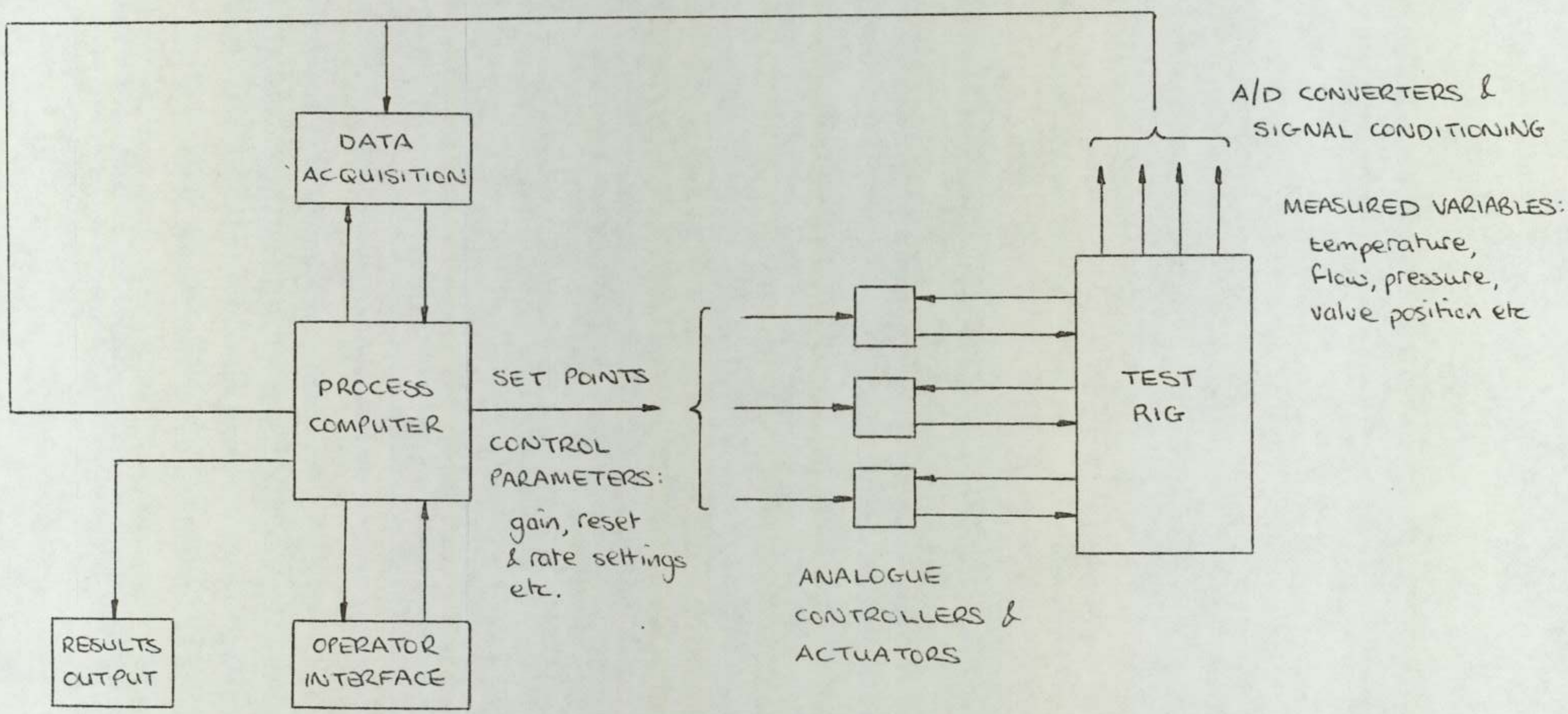


FIG. A6.12 HYBRID CONTROL APPLIED TO THE AUTOMATED TEST RIG.

illustrates hybrid control applied to the automated test rig.

A6.7. APPROACHES TO THE DEVELOPMENT OF AN AUTOMATED TEST RIG

This Appendix has highlighted areas where the company is lacking in technical know-how and experience, particularly in those of instrumentation and control. The first step towards automation should be to remedy this situation. Much time and effort can be saved by speaking to equipment manufacturers, research organisations, consultants, etc., with experience in these fields rather than "going it alone" from the outset. Some preliminary temperature measurement proving tests may well be necessary so that first-hand experience and confidence can be obtained gradually, for example, to compare directly temperature measurement by mercury in glass thermometers and thermocouples. Also, the magnitude of the control problem, namely temperature control, can be quantified and qualified by some initial experimentation.

There are many possible engineering solutions to the problem of automatic testing, resulting in various levels of automation and hardware requirements. Perhaps the ultimate restrictions to full automation will be cost and cost justification. Once a realistic measure of the control problem has been obtained, then one approach towards a solution would be to carry out an equipment survey to establish what equipment is readily available, its applicability to the test requirements and its limitations. The most sophisticated and powerful equipment in the range of process controllers are the process control computers, of which several makes are on the market. They are expensive, £20,000+ for the computer system alone. One application of process control computers is the automatic control of chemical plants and consequently their full capability may be under-utilised for the test rig application. In addition, to quote a well known phrase, "this could be a case of using a sledge-hammer to crack a nut". A second approach could be to develop ones own specialised controllers based on microprocessor technology when suitable control equipment is not available. Disadvantages of this approach are that the necessary expertise is not available within Serck Heat Transfer, extensive development and technical backup facilities are required, software development takes a long time and is expensive, and all work would need to be carefully documented in the event of the designer(s) leaving the company. Only after an initial study of the whole problem, including the various options for automation is completed, can a realistic and accurate design specification be written.

A development approach which may be favoured is to divide the development of the automated test rig into a series of distinct operational stages, each progressing to a higher level of automation than the previous stage until ultimately a fully automated test rig is developed, provided, of course, the latter can be justified. It is possible that the company may be satisfied with a part automated rig and so full automation will not be achieved. To ensure that the various stages of part automation can each be extended to the next without too much difficulty, it is essential to design the test rig for flexibility. This requires a clear understanding of what automation involves, how it can be accomplished, and designing the test rig and specifying equipment for the initial development phases which will be compatible with future requirements. As has been indicated, there will probably be a number of alternative options in the design, however, it is the common features which need to be identified so that a flexible development programme can be drawn up.

An alternative approach to the problem of assessing the wide and unknown variation in the control parameters and the merits of different control strategies would be to carry out a computer simulation. However, before embarking on this course, the time, effort, expense, and real value of the exercise would need to be justified.

A6.8. CONSEQUENCES OF AUTOMATED TESTING

In this section, some of the consequencies of introducing an automated test facility into the company's laboratory are raised.

Table A6.3 gives an estimate of the time taken and the running costs for a typical thermal performance test carried out on the present 3 inch test rig.

An estimate for the cost of an automated test rig is presented in Table A6.4. A fully automated test rig can be expected to cost upwards to about £100,000. This estimate is for the capital cost of the test rig hardware only and therefore excludes the costs associated with development, installation, operating overheads, and technical back-up facilities.

Introducing high technology equipment into the laboratory has certtain implications for the company. An automated test rig needs to be maintained and run, therefore it is essential to provide good technical back-up facilities. These would include calibration facilities for temperature,

TABLE A6.3. TIME & COST ESTIMATES FOR A TYPICAL OIL/WATER THERMAL PERFORMANCE TEST CARRIED OUT ON THE EXISTING 3 INCH PLANT

Piping up oil and water circuits and fitting test unit	5 days
Instrumentation	1 day
Water Isothermal pressure loss tests	½ day
Oil Isothermal pressure loss tests, 2 temperatures/day	say 2 days
Heat transfer tests, 30 test points, at 5 points/day	6 days
Remove test unit from rig and break down pipework	1½ days
	16 days
+ 25% for contingencies	4 days
	20 days
cost at £80.00/8 hour day = £1600.00	

(Costed at September 1979)

TABLE A6.4. ESTIMATE OF THE HARDWARE COSTS FOR AN AUTOMATED TEST RIG

QTY	DESCRIPTION		TOTAL PRICE, £
6	Temperature sensors and transmitters		1815.00
2	Temperature controllers (P + I + D)		1341.00
1	Lead/lag unit + computing elements		1520.00
6	Turbine meters (oil) and transmitters		4347.00
3	Magnetic flow meters (water) and transmitters		3972.00
3	Steam orifice plates, transmitters and square root extractors		2467.00
4	Differential pressure transmitters		1656.00
2	Differential pressure controllers (P + I)		1265.00
4	Flow controllers (P + I)		2696.00
6	Oil circuit control valves		2648.00
6	Water circuit control valves		3011.00
3	Steam circuit control valves		1321.00
	Alarms and accessories		4309.00
			32368.00
	Data logging system	assume	6000.00
	Steam heaters, pumps, pipework, etc.	assume	15000.00
			53368.00
	Process computer	assume	30000.00
			83368.00
		say, up to	100000.00

(Costed at September 1979)

flow, and pressure measurement, and the general maintenance of the electronic equipment, although some calibration may be carried out at specialist calibration centres. Additionally it is important to have the right calibre of personnel for the job. This would necessitate training the existing laboratory staff and/or employing suitably qualified technicians or engineers.

In the future, the possibility will exist for extending automation to other test areas which would benefit, for example, the plenum chamber test rig for performance testing of radiator type heat exchangers. And perhaps eventually, this could lead to a general data acquisition and processing facility for the laboratory.

A6.9. CONCLUSIONS

1. How the control parameters, which define how the test rig will respond to set point and load changes vary and to what extent is unknown. The complexity of the control problem will significantly influence the design for an automated test rig. This preliminary assessment seems to indicate that feed forward control is appropriate, however, will less sophisticated control techniques prove adequate?
2. Automatic data logging equipment can record measurements almost instantaneously; therefore, the data collection time can be reduced and for practical purposes the effect on the results of changes in process conditions during recording of measurements can be eliminated. The possibility then exists to collect and analyse more data in a given time, particularly quasi steady state data. However, how steady the process conditions need to be in order to give reliable results will need

to be quantified.

3. The company's expertise in the use of transducers whose outputs are suitable for an automated test rig is limited. In particular, confidence needs to be gained in temperature measurement techniques.

4. At present the company does not have the technical knowledge and experience, nor suitable qualified personnel to develop an automated test rig.

A6.10. RECOMMENDATIONS

When the company has indicated its desire to reduce testing time by automation of the conventional steady state method, an in-depth feasibility study should be undertaken to:

1. Qualify and quantify the control problem with reference to:
   (a) The control strategy necessary for satisfactory control of the process variables, temperature and flow.
   (b) Sequencing of test conditions; adjustment of the set points and the controller parameters; the start-up and shut-down procedures.
   (c) The steam capacity required to "drive" the rig to the required steady state test condition.
2. Quantify what is an adequate steady state condition.
3. Investigate how the various automation schemes can be achieved, i.e., the hardware requirements.
4. Consider the implications of automated testing with regard to the technical back-up facilities and the calibre of personnel required.
5. Consider how the schemes for automated testing fit in with the anticipated future testing requirements of the

laboratory as a whole.

6. Use the results of 1-5 to define as precisely as possible the complete specification for an automated test rig.

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