HEAT LOSSES IN INTERNAL COMBUSTION ENGINES

by

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September 1989

A dissertation submitted to the University of Cape Town in fulfilment of the requirements for the degree of Master of Science in Engineering.

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ABSTRACT

This thesis deals with the effects of cooling and heat losses in internal combustion engines. The object of this work was to examine and research various cooling concepts and methods to reduce heat loss to engine coolant, improve thermal efficiency and to predict heat transfer values for these alternatives. The optimum system to be considered for possible application to small rural stationary engines.

A literature survey was undertaken, covering work performed in the field of internal combustion engine cooling. Besides the conventional cooling system, two concepts emerged for consideration. These were the precision cooling system and the new heat pipe concept, the latter being relatively unknown for internal combustion cooling application.

The precision cooling system, consists of a series of small bore tubes conducting coolant only to the critical areas of an engine. The theory being that in the conventional systems many regions are overcooled, resulting in excessive heat loss.

The heat pipe is a device of very high thermal conductance and normally consists of a sealed tube containing a small quantity of fluid. Under operating conditions the tubular container becomes an evaporator region in the heat input area and a condenser region in the heat-out area. It is therefore basically a thermal flux transformer, attached to the object to be cooled. The heat pipe performance is also capable of being modulated by varying its system pressure. This is a positive feature for internal combustion engine application in controlling detonation and NO_x emissions.

Various facts were obtained from the literature survey and considered in the theoretical review. These facts were extended into models, predicting the heat transfer performance of each concept in terms of coolant heat outflow and heat transfer coefficients.

The experimental apparatus was based on an automotive cylinder head with heated oil passing through the combustion chamber and exhaust port to simulate combustion gases. Experiments were conducted on this apparatus to validate the predicted theoretical performance of the three concepts. Tests were also made to observe the effect of heat pipe modulation and nucleate boiling in the precision system.

Concept theory was validated as shown by the experimental and test results. The performance for each system approximated the predicted heat transfer and heat loss values.

By comparison of the heat input, coolant heat outflow values and heat transfer coefficients it was found that the precision system was the most efficient, followed by the heat pipe and the conventional system being the least efficient.

It was concluded that the heat loss tests provided a valuable insight into the heat transfer phenomenon as applied to the three systems investigated. This work also illustrated the effects of the variation of coolant flow, velocity and influence of nucleate boiling.

This thesis has shown the potential of the systems tested, for controlling heat losses in internal combustion engines. The research work has created a data base for further in-depth evaluation and development of the heat pipe and the precision cooling system.

Based on the findings of the experimental work done on this project, several commercial applications exist for the heat pipe and precision cooling systems. Further in-depth research is recommended to extend their potential in the automotive industry.

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NOMENCLATURE

L	Latent heat of vaporisation
ρ_l	Liquid density
g	Acceleration due to gravity
K	Wick thermal conductivity - liquid phase
μ	Dynamic viscosity of liquid
ℓ_c	Condenser length
ℓ_a	Adiabatic length
ℓ_e	Evaporator length
μ_l	Liquid viscosity
D	Pipe diameter
Pcm	Capillary pressure
σ	Surface tension coefficient
r_c	Capillary radius
$(QL)_c$	Heat transport factor
$\Delta P_{\perp}$	Hydrostatic press in direction perpendicular to pipe axis
Lt	Total length of heat pipe
ψ	Heat pipe inclination from horizontal position
F_l	Frictional coefficient for liquid flow
F_v	Frictional coefficient for vapour flow
A_w	Wick cross-sectional area
Q	Heat transport - watts
Tv	Vapour temperature

GLOSSARY

AXIAL HEAT FLUX: In a cylindrical heat pipe, is the axial heat transfer along the length of the evaporator to the condenser region.

BLANKETING: Occurs under boiling liquid conditions when continuous vapour forms as a result of high bubble population, and liquid has difficulty in reaching the heated surface.

CONVENTIONAL COOLING: Existing automotive cooling system with water flowing through water jacket in cylinder head.

FLUID INVENTORY: The quantity of liquid in the heat pipe.

GAS BUFFERING: Method of varying heat pipe effective condenser area, according to change in thermal input and maintaining constant pressure.

GRAVITY ASSISTED HEAT PIPE: A heat pipe with the condenser elevated above the evaporator, allowing condensate to return to the evaporator by gravity.

HEAT PIPE: A high thermal conductance device consisting of an evaporator, condenser and a capillary wick.

HEAT PIPE MODULATION: Variation of thermal conductance by changing the vapour pressure in the heat pipe.

INTEGRATED HEAT PIPE: Experimental heat pipe using a cylinder head water jacket to simulate the evaporator.

ISVR: Institute of Sound and Vibration Research, University of Southampton.

NUCLEATE BOILING: When a liquid is heated to a temperature beyond its boiling point.

NUSSELT THEORY: Theory of forced convection heat transfer - identified by the Nusselt number and appropriate equation.

POOL BOILING REGIME: Is the pattern of heat flux of a liquid, developed as the heating surface temperature is increased from boiling to beyond the liquid boiling point.

PRECISION COOLING SYSTEM: A system developed to cool only critical areas of an engine by means of small bore coolant tubes.

SMALL BORE SYSTEMS: Small diameter coolant tubes used in the precision system.

SONIC CHOKED FLOW: Occurs when the heat pipe condenser temperature is lowered and sonic velocity is achieved at the end of the evaporator.

THERMAL FLUX TRANSFORMER: A heat pipe characteristic, being able to take in heat at high flux and heat out at low flux.

THERMAL SYPHON: A heat pipe without a capillary wick.

VAPIPE: A heat pipe used to vaporise fuel by transforming heat from the exhaust to the inlet manifold.

VARIABLE CONDUCTANCE HEAT PIPE: A modified heat pipe, providing temperature control by modulating the vapour pressure within the pipe.

WICK: Normally a wire mesh to provide capillary action for the return of heat pipe condensate to the evaporator.

CHAPTER ONE

INTRODUCTION

The title for this project was based on a preliminary examination by the Energy Research Institute of various energy and efficiency concepts associated with the design consideration for a rural engine application.

South Africa's population is expected to expand significantly by the turn of the century. Whilst most of the country's growth will be concentrated in the metropolitan areas, a corresponding increase in rural development must take place.

This rural expansion, together with increased agricultural activity, will result in the need for self-contained power generators and prime movers. It is anticipated that the small cheap direct power unit, preferably a multifuel unit, will be the primary source of power for water pumping, electricity generation, workshops and hospitals.

A recent survey carried out by the Energy Research Institute indicates that the most popular engines are small units with an engine output of approximately 8 kw; these engines being fully imported except for one unit which is partially locally manufactured. Costs vary from R3 600 to R6 000 (1988) and designs are dated.

Because of these factors a project was motivated to investigate efficiency, concept and design improvements for a potential locally manufactured power unit to meet the above applications.

It was felt that engine prices would be significantly improved with a locally designed and manufactured engine, thereby stimulating local production. In addition, the current engines available could possibly be improved especially in the areas of mechanical simplicity and thermal efficiency.

With a more modern design, the specific fuel consumption could also be reduced, therefore placing emphasis on the cooling system and potential reduction of heat losses. Where a temporary fuel scarcity exists due to supply or cost problems, an engine with multifuel capability would be desirable and a cooling system capable of optimising engine efficiency by reducing heat losses is essential.

The reduction of heat losses to the coolant can be achieved by cooling only the critical areas of the engine, such as the valve guides, injector seals, valve bridge and upper part of the cylinder liner. The principle of small bore water passages and high fluid speeds could be employed to obtain high rates of heat transfer in these areas. The use of high cooling rates only where required, will not only reduce heat losses from other areas, but will also reduce thermal stresses in the critical areas.

Normally, water cooling is the more widely adopted method for cooling internal combustion engines, one of the alternatives being air.

The basic requirements of a water cooling system to control heat losses are:-

(i) The metal surfaces bounding the combustion chamber must be held at a temperature below that at which deterioration of the crystal structure, oxidation or other processes occur.

(ii) The temperature differences throughout the metal structure must be such as to avoid excessive thermal stresses.

(iii) The temperature of lubricated surfaces, especially the cylinder bore and the valve guides, must be kept low enough to ensure adequate life for the lubricating oil and ring grooves.

(iv) In the case of diesel engines, the fuel injection nozzle must be kept cool enough to avoid nozzle hole blocking by carbon formation or needle valve sticking by lacquering of the needle valve stem.

THE SCOPE OF THIS PROJECT IS THE COMPREHENSIVE EXAMINATION, STUDY AND RESEARCH OF THE VARIOUS ALTERNATIVE CONCEPTS AND METHODS WHICH COULD BE EMPLOYED TO REDUCE HEAT LOSSES IN INTERNAL COMBUSTION ENGINES.

CHAPTER TWO

LITERATURE SURVEY

This chapter concerns the effects of various cooling concepts and the phenomena of heat losses in internal combustion engines. Many researchers have covered the topic of cooling, heat transfer and losses and contributed significant work into the understanding of engine cooling and heat control. This has come about by the continued strive for improvement of internal combustion engine efficiency and latterly the side effects of detonation and emission control of vehicles.

Thermal losses in an internal combustion engine are always significantly more than the mechanical losses. This suggests that the main thrust to improve fuel efficiency should be through reduced heat to coolant or exhaust or by combustion efficiency.

2.1 EFFICIENCY AND HEAT BALANCE

A typical direct injection diesel engine, operating at maximum power and speed, according to Holmes[4], will convert approximately 36% of the available fuel energy into power.

Approximately 22% of the remaining 64% is passed into the engine's cooling system, and the remaining 42% is used to drive vehicle auxiliaries, or escapes through the exhaust.

This suggests that smaller cooling systems can be specified, if the amount of energy lost can be reduced, more power will be available with the changing energy balance or hotter exhaust. Engine efficiency will be further increased by the greater exhaust energy.

Holmes(4) conducted engine simulation studies, using a practical engine design, with reduced losses and an exhaust recovery turbine. This gave the heat balance figures shown in Fig. 1.

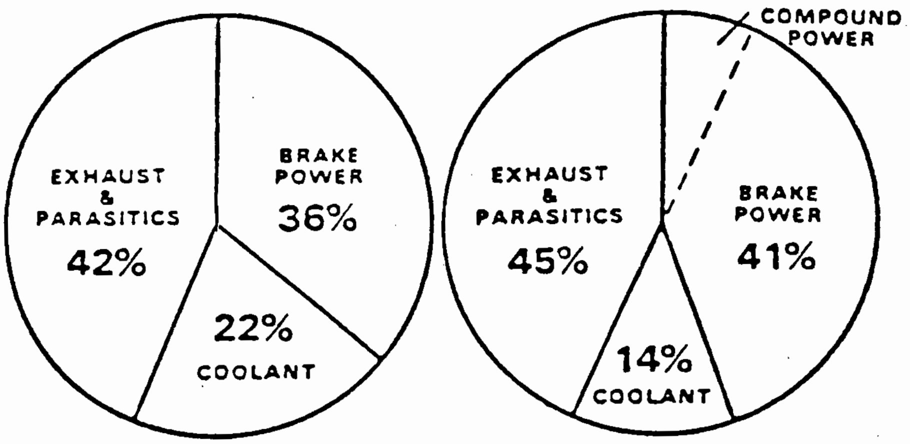


Fig. 1. Energy balances for turbocharged and turbocompound diesel engine.

This indicates a calculated 5% increase in engine efficiency which equated to more than 13% improvement in fuel economy.

A further investigation to reduce the heat loss to coolant in a practical diesel engine was to minimise the amount of cooling water in the engine, and allowing metallic components to run hotter. Precision cooling of critical areas of the engine, such as the cylinder head valve bridge and injector sleeve, was used to ensure reliability.

The computer engine simulation model used showed that heat losses within a standard engine are generally equally distributed between the pistons, cylinder head and cylinder block, as shown in Fig. 2.

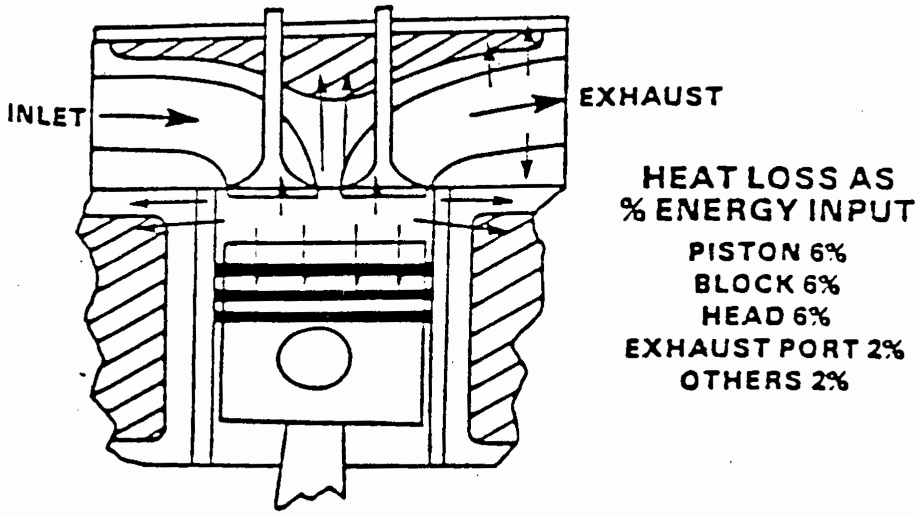


Fig. 2 Heat flow in a standard diesel engine

This diagram illustrates the importance of being able to cut the heat loss through all in-cylinder components. However, the effect of insulating one component only, such as pistons, is to increase temperatures and stresses within other parts of the engines.

In addition to the simulation model used, further work on optimisation of cylinder head water flows was carried out using an acrylic model of the head attached to a flow rig, enabling high speed photography of the water flow to show up any areas of cavitation or stagnation. Temperatures measured within the head were found to match the temperatures predicted by the finite element model, thus giving confidence in the head design and the modelling techniques used.

Heat flow through the head was found to be considerably reduced by the precision cooled design, and exhaust temperatures increased, illustrating that the engine's energy balance had been altered.

These findings were used as a basis for establishing the extent of heat flow from a typical cylinder head and the values applied to the theoretical and experimental considerations of this project.

2.2 ENGINE COOLING

To fully understand the heat loss phenomena of internal combustion engines and various ways of minimising heat loss and increasing efficiency, we need to understand and examine the restrictions placed on cooling systems by the physical properties of the coolant. This is especially applied to water which is the natural choice for an engine coolant.

For example, a spark ignition engine has to be retarded to avoid detonation when water coolant temperatures rise above a relatively low level, thus limiting power and fuel economy. Maintaining coolant temperatures low enough to prevent loss of power, limits the size of cooling fans and radiator.

Critical areas of the engine that receive large quantities of heat experience localised boiling of the coolant. In confined spaces around the exhaust valve seats for example, coolant will boil under wide-open throttle and high load conditions, regardless of the bulk temperature or flow velocity of the coolant according to Evans and Right[6].

2.3 RESPONSE TIME OF A MODULATED ENGINE COOLING SYSTEM

According to Priede[18], the rapidity with which an engine throttle may be opened during driving makes rapid readjustments of the metal surface temperatures necessary.

One of the objects of testing the various cooling/heat loss concepts is to investigate the highest possible rates of heat transfer, and this will greatly assist in minimising response time in proposing a future cooling modulation system.

2.4 NUCLEATE BOILING

When coolant liquid is in direct contact with metal that has been heated to a temperature beyond the boiling point of the liquid, the boiling regime is termed NUCLEATE. Where the nucleate phenomena exists, the heat transfer at the junction of the metal and the liquid is enhanced by agitation at the junction.

Mansfield and Grover[11] amplify this by stating that the results of a large number of research papers clearly show that among the many hypotheses, the most appropriate seem to be those assuming that the vapour bubbles during their departure entrain in their wake a part of the superheated boundary layer, transporting it into the cooler bulk of the liquid while the cold liquid flows back into the cavity and is heated by transient heat conduction.

Mansfield quotes that Fishenden and Saunders in their "An introduction to heat transfer" state that chains of bubbles rise through the liquid, their sweeping and stirring action producing a greatly increased heat transfer.

Their references also state that the volume of boundary layer torn off is equal to about one half of the volume of the bubbles. The ratio between water density and steam density is:

$$\frac{62}{0,036} \quad \text{i.e. } 1720.$$

Each gram of water converted to steam removes 539 calories and occupies 1720 times its original volume. The volume of boundary layer torn off is therefore $\frac{1720}{2}$ ml.

If the mid stream temperature is 80°C, the heat transferred by removing the boundary layer at 100°C and replacing it by water at 80°C is 860 x 20 calories, i.e. 17200 calories compared with the 539 calories carried away in the steam, i.e. 32 times as much. This calculation neglects the superheating of the boundary layer which increases the heat it takes with it.

An in-depth review of nucleate boiling is relevant as some of the associated factors have a bearing on the experimental results obtained for this project.

According to Dunn and Reay[1], the mechanism of bubble formation depends strongly on the **WETTING** characteristics of the heating surface. For a bubble to form it must start at a nucleation centre which provides a finite initial radius. In addition the liquid must be superheated in order to provide the pressure difference. The effect of wetting on bubble formation is shown in Fig. 3(a).

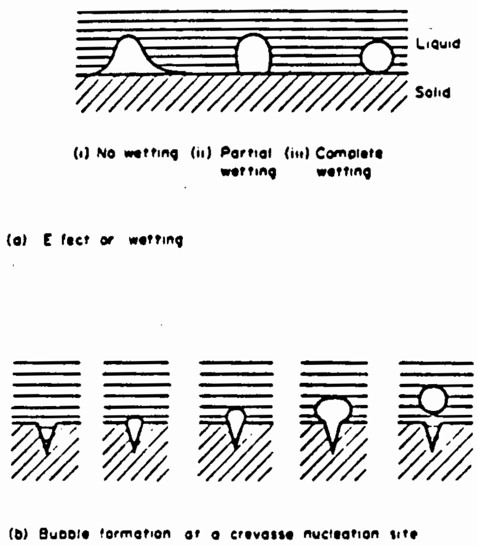


Fig. 3. Bubble formation on a heated surface.

Bubbles form most readily if the surface is non-wetting. In addition to wetting, Nucleation sites are necessary for bubble formation.

Nucleation sites are provided by scratched or rough surfaces and by the release of absorbed gas.

Fig. 3(b) shows how a bubble forms in a crevasse in a surface and is the condition which could be expected in the water jackets of cylinder heads. As might be expected, a much higher superheat is required to form bubbles on a clean, smooth surface.

Fig. 4 shows how the temperature varies with distance from the surface under nucleate boiling conditions.

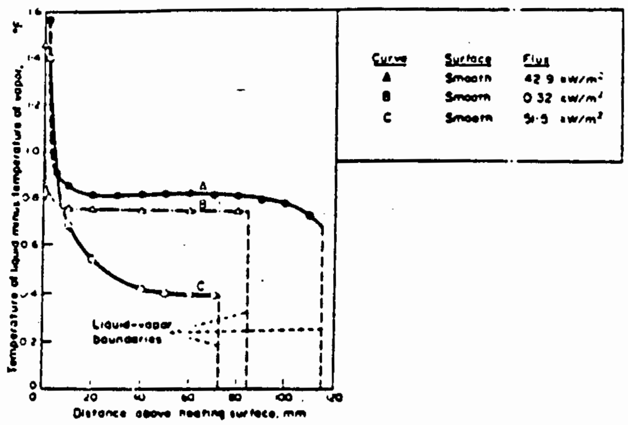


Fig. 4 Variation of temperature with distance from a surface under nucleate boiling considerations.

Nucleate boiling is very dependent on the heated surface and factors such as release of absorbed gas, surface roughness, surface oxidation and wettability, greatly affect the surface to bulk liquid temperature difference. The nature of the surface may change over a period of time - a process known as "conditioning". For these reasons reproducability of results is often difficult.

For example, in certain combinations of stainless steel and sodium it was found that due to the wetting of the metal by the sodium, nucleation was difficult to start and a superheat of 100°C was required, dropping to 20°C after the surface became conditioned.

Nucleate boiling, however, complicates the mechanisms of heat transfer and heat transfer coefficients due to the complex geometry of the coolant flow passages in most engines.

As long as the coolant liquid directly contacts the coolant jacket metal, the metal surface temperature can rise only slightly above the boiling point of the coolant, regardless of the heat flux at the metal coolant interface.

Thus critical metal temperatures in engines are limited by the boiling point of the coolant and it is essential that conditions promoting nucleate boiling are maintained to prevent hot spots from forming in engine cylinder heads.

"Hot spots" are formed by sudden increase in metal temperature as a result of the jacket metal becoming insulated from the coolant by a vapour layer. Under these conditions the relationship of the metal temperature and the boiling point of the coolant is severed and "blanketing" is said to have occurred.

2.5 WATER AS A COOLANT

A disadvantage of water, when used as a coolant for engines, is that water vapour condenses at a relatively low temperature.

According to Evans[6] water vapour generated by boiling in the cylinder head must condense into the coolant itself. Under heavy load and high ambient temperature conditions, when the coolant temperature approaches the saturation temperature of water, the vapour in the cylinder head cannot condense soon enough to prevent its occupying and insulating critical areas of the cylinder head. Hot spots resulting from the displacement of liquid coolant away from the hot metal cause detonation and excessive NO_x emissions.

Therefore, to prevent failure of a conventional cooling system, it is vital that the bulk coolant temperature in the cylinder head never exceeds the temperature at which water vapour will condense. The saturation temperature of water vapour will be the failure point of the cooling system. If this limit is breached, none of the vapour generated can be condensed and a major portion of the coolant in the coolant jacket is expelled.

CONSIDERATION OF VARIOUS COOLING SYSTEMS

2.6 PRECISION COOLING

One of the objectives of this project was to research the effects of precision cooling by using small bore high velocity cooling systems.

The ISVR Division of Southampton University has done extensive research on the reduction of coolant losses in internal combustion engines and developed this unique cooling concept in conjunction with industry.

Over the last decade Dr W P Mansfield(7) of ISVR investigated and applied a design philosophy of cooling only the critical regions of the engine by small bore cooling passages. It was found that satisfactory working temperatures could be obtained with considerably reduced heat removal.

Studies of actual engines show a wide variation of temperature across engine surfaces where the maximum permissible temperature is constant. For example, temperatures across the combustion face of a cylinder head may vary from 100°C to 400°C. If there are areas of the combustion chamber or cylinder bore surfaces which are cooler than the highest local permissible temperature, then the heat transfer from the combustion gases to these surfaces both by convection and by radiation will be greater than the possible minimum.

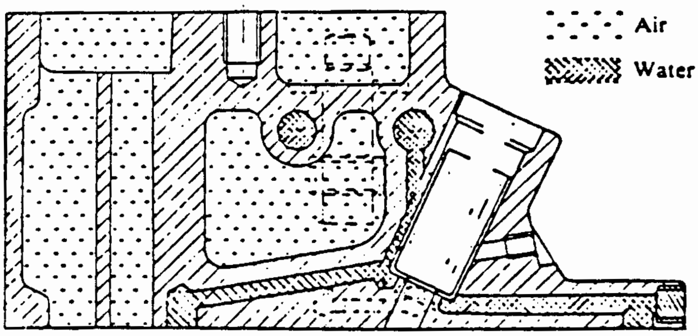
Research done in the UK by Mansfield[7] and Priede[18] at the Institute of Sound and Vibration Research (ISVR) (University of Southampton) has shown that cooling of the critical areas only, e.g. valve bridge, injector housing, exhaust port and the cylinder liner and head deck can adequately control internal combustion engine cooling. Any additional cooling only absorbing energy and reducing engine efficiency.

This research work indicated that cooling of the critical areas could be achieved by small bore precision passages, located in these areas, resulting in satisfactory homogenous working temperature and a substantial reduction in the flow of heat to the cooling water.

The benefits of precision cooling are: improvements in engine performance, reduction of thermal stresses, reduction of fan noise, increased exhaust energy for turbo-charging and compounding and ease of casting are all obtained by changing the cylinder head without the use of insulating materials.

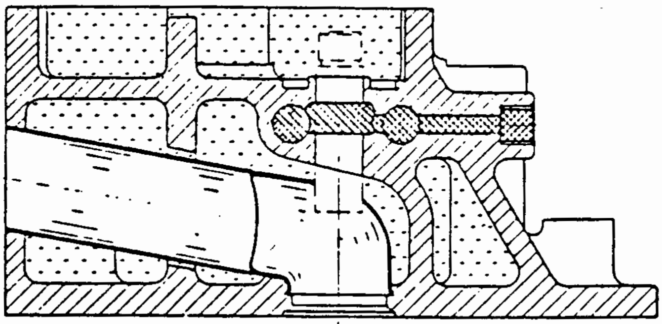
The rationale for the precision cooling system is that for conventional engine designs, the engine cylinder and cylinder cover are surrounded by box-like casings, with intervening spaces forming the water passages. The cylinder casing, walls and upper deck of the cylinder head are also used as load carrying components. These are not present in air-cooled engines, indicating that they are not essential for this purpose. Where a higher rate of heat flow through the walls of the cylinder and head is required to maintain metal temperatures, this is often achieved by locally reducing the area available for water flow so that relatively high velocity flow over critical surfaces is obtained. Present cooling systems, therefore, may be effective but their efficiency is in question.

The basic principles of the Southampton University ISVR precision cooling system are illustrated in Fig. 5 and Fig. 6 as applied to the Perkins 4.236 diesel engine cylinder head.



Schematic diagram of precision cooling system applied to diesel cylinder head

Fig. 5



Schematic diagram of precision cooling system applied to diesel cylinder head showing insulated exhaust passage

Fig. 6

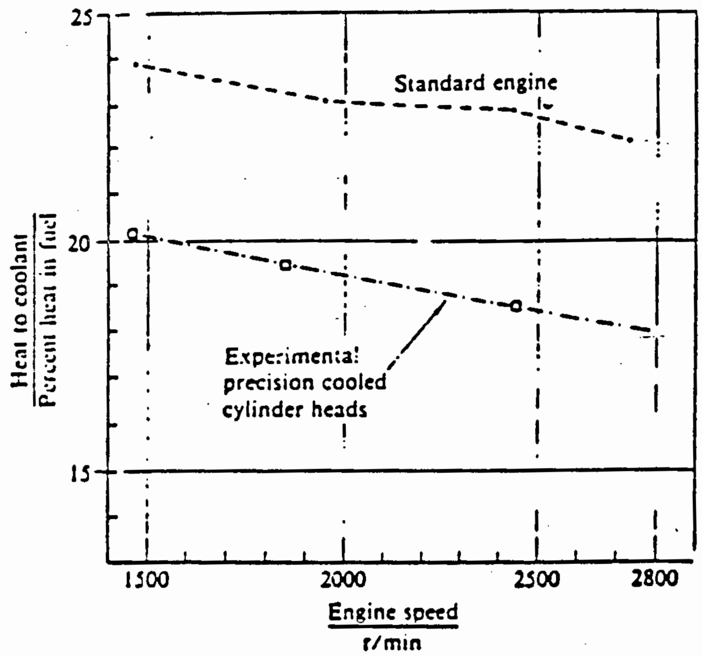
Most of the space between the two decks is occupied by air which previously was filled with cooling water. The illustration shows that small local passages are used to cool critical regions only, such as the injection nozzle, valve bridge and exhaust valve seat regions in the lower deck and the valve guides just below the upper deck of the cylinder head.

This system allows the coolant flow to be fully controlled at suitable velocities through the use of suitably designed small bore local passages. By controlling the temperatures in the critical areas of the cylinder head, reliable operation without unnecessary heat removal can be achieved.

Development work has shown a significant aspect of precision cooling is that other areas of the cylinder head which are not directly cooled will experience a rise in temperature above former values but remain well within safe limits. Lower thermal stresses are developed due to the more uniform distribution of temperature.

> **BECAUSE UNNECESSARY REMOVAL OF HEAT IS AVOIDED PRECISION COOLING PRODUCES A REDUCTION IN THE FLOW OF HEAT TO COOLANT.**

Fig. 7 shows the results obtained by the ISVR research programme.



Reduction in heat to coolant achieved by precision cooling head on a 3.9 litre diesel engine

Fig. 7

The change to the PRECISION COOLED HEAD resulted in a reduction of about 12% of the original heat flow to coolant at full load at 1500 rpm and a reduction of about 18% at full load at maximum rated speed of 2800 rpm.

The reduction of heat flow to coolant of 10% realised a reduction of required cooling fan power by 40% to 60%.

2.7 COOLANT FLOW CHARACTERISTICS AND HEAT TRANSFER IN SMALL BORE SYSTEMS

Investigations have been made on a test rig of heat transfer coefficients employing coolant flow in small cross sectional areas.

It was found that with a 5 mm diameter passage and a water velocity of 5 to 8 m/sec, heat transfer coefficients of 3,6 to 5,3 w/cm^2 ^oC were obtained. Finlay[5] confirmed these findings but with lower coolant velocity of 1 m/sec.

These results will be the subject of further investigation during the theoretical development and experimental procedure chapters of this thesis.

The above heat transfer rates are comparable with those which can be obtained by nucleate boiling and safer than uncontrolled nucleate boiling. ISVR[10] suggests that the Nusselt equation indicates an increasingly rapid increase of heat transfer coefficient as the passage area is reduced. This being explained by the possible absence of nucleate boiling in very small cross-sectional area passages being due to very high rates of heat transfer by forced convection and the high rate of heat removal from the boundary layer, preventing boiling.

Additional observations were advanced by Mansfield[11]. Heat transfer from the walls of the cylinder head to the coolant is assumed to take place by a combination of forced convection and sub-cooled nucleate boiling. He suggested the following method for determining heat flux when the mass flow of vapour is very small or zero:-

$$\text{Nusselt equation} = 0{,}023 \left(\frac{\rho v d}{\mu}\right)^{0,8} \times \left(\frac{c\mu}{k}\right)^{0,4}$$

$$\text{i.e.} \quad \frac{hd}{k} = 0{,}023 \left(\frac{\rho v d}{\mu}\right)^{0,8} \times \left(\frac{c\mu}{k}\right)^{0,4}$$

where:-

h = coefficient of heat transfer — w/sq.cm^{o}C

d = passage diameter — m

k = thermal conductivity of liquid — w/m^{o}C

ρ = density of liquid — kg/m^{3}

v = velocity of liquid — m/sec

μ = viscosity of liquid — kg/m/sec

c = specific heat of liquid — KJ/kg^{o}C

The above equation will be used in the subsequent chapters of this thesis to establish the calculated heat transfer values using the experimental coolant velocities. The heat transfer values achieved experimentally will then be compared with the calculated values.

This project, titled "HEAT LOSSES IN INTERNAL COMBUSTION ENGINES" will re-examine the previous work done by ISVR Southampton University, on the precision cooling system. This will be compared with two other cooling systems. These are the Heat Pipe Concept, as an internal combustion engine cooling medium and the conventional automotive cooling system. The latter being used as a basis for comparison of the experimental cooling and heat loss results.

The parameters of the precision cooling concept will be applied to the Heat Pipe evaluation viz optimum cooling of the critical engine components only.

2.8 HEAT PIPES

Initially the main thrust of this project and subject of this thesis was to extend the development work done on precision cooling and to prepare apparatus similar to that used by Southampton University to develop the precision cooling concept using local criteria. However, practical problems with the proposed apparatus dictated that a new test rig concept would be required. These problems will be reviewed under "EXPERIMENTAL PROCEDURE", chapter 4.

Before finalising the experimental rig for precision cooling the opportunity was taken to consider other methods available to examine heat losses in internal combustion engines. Where an alternative method could be identified, then ideally a common apparatus and test procedure be developed to suit both the alternative concept and precision cooling research.

As detailed earlier in this report much research has been conducted on conventional cooling systems and the results well documented. A new approach to establishing engine heat loss and cooling was, therefore, considered desirable for this project. This would not only initiate new research but also establish basic data for others to move forward confidently in expanding the new concept. This data would be used for comparison with the precision cooling and conventional system heat transfer data obtained during the experimental work on this thesis.

After considering possible alternatives it was decided to investigate the new HEAT PIPE concept.

A literature survey revealed that virtually no information existed on heat pipe application for automotive purposes. Indications are that heat pipes are mainly being applied to solar receivers, recovery of waste heat from industrial processes, conservation of heating energy, electrical componentry and space craft equipment.

The only automotive reference found was by Harner J L[8] and Lindsay and Wilson[9]. Harner describes two design concepts for using heat pipes in an early fuel vaporisation application. Lindsay and Wilson introduced a heat pipe device with a number of unique characteristics which made it a particularly appropriate means of transferring exhaust heat to vaporise gasoline. They claimed the concept has an effective thermal conductance of 500 or more times that of copper.

By this means heat from the exhaust is transferred to the fuel air mixture via the induction manifold causing fuel evaporation. Under these conditions a mixture as lean as 22:1 will ignite without difficulty, resulting in reduction in NO_x and CO levels.

Fig.8 shows a diagram of the vapipe's position and application.

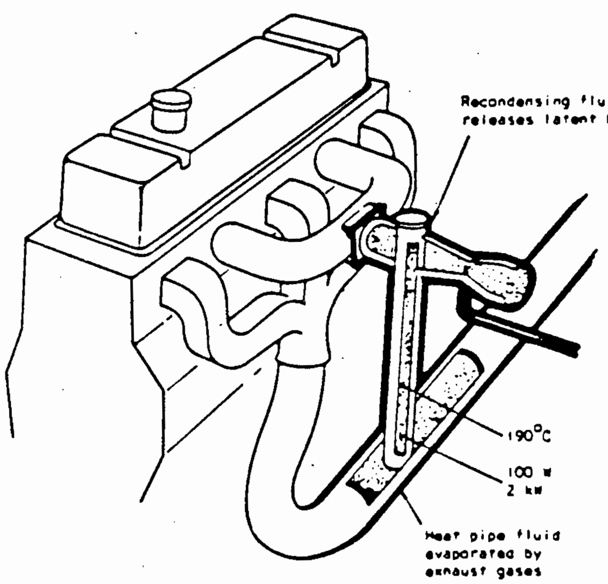


Fig. 8 Location of Vapipe

Reference is made here to the Vapipe, illustrating the feasibility and practical application of a heat pipe for automotive use. The heat pipe application for cooling internal combustion engines will, however, be unique and no literature on this application appears to be available or mentioned by heat pipe authorities such as Dunn and Reay[1] and CHI[2].

Lindsay and Wilson[9] draw the conclusion that much work still has to be done to take the Vapipe from the current development stage to its use in the everyday motor car. Much of it inevitably will lie in the province of the automotive industry but the opportunities offered are sufficient to make this concept attractive.

This project therefore initiates new research in the cooling and heat transfer performance of heat pipes as applied to internal combustion engines.

For a meaningful later discussion and analysis of the experimental results it is necessary to examine the heat pipe theory in detail and to explore its potential for cooling internal combustion engines, thus establishing data to determine heat loss and potential effect on efficiency and economy on engines.

2.9 HEAT PIPE SYSTEM

The heat pipe was first suggested by R S Gaugler in 1942. It was not, however, until its independent invention by G M Grover in the early 1960's that the properties of the heat pipe became appreciated and serious development work took place.

According to Dunn and Reay[1] the heat pipe is a device of very high thermal conductance, that is, it will transport thermal energy without an appreciable drop in temperature. The heat pipe may be regarded as a development of the thermal syphon and it may be helpful to describe the operation of the latter before discussing the heat pipe.

The thermal syphon, Fig 9(a) consists of an evacuated sealed tube containing a small quantity of liquid.

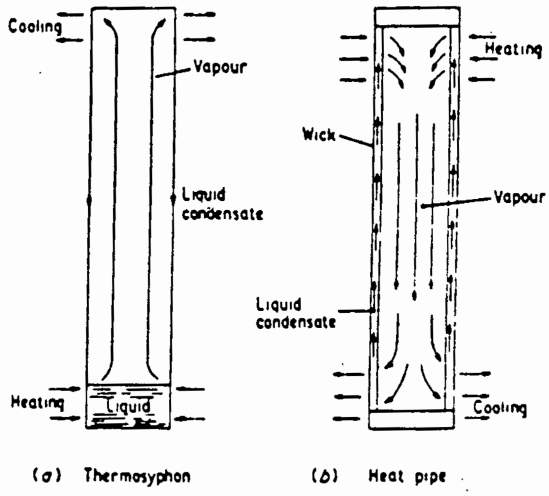


Fig. 9

If this liquid at the bottom of the tube is heated, it will evaporate and the resulting vapour will flow to the top of the tube, where it will condense, returning to the evaporator region by gravitational forces. The net effect is for thermal energy to be transported from the evaporator to the condenser end of the tube.

Since the latent heat of evaporation is large, considerable quantities of heat can be transported with a small temperature difference from end to end. Thus the structure will have a high effective thermal conductance.

The thermal syphon has been used for many years and various working fluids have been employed. One limitation of the basic thermal syphon is that in order for the condensate to be returned to the evaporator region by gravitational force, the evaporator must be situated at the lower point.

The heat pipe, Fig. 9(b) is similar in construction to the thermal syphon but in this case a wick, constructed for example from a few layers of fine gauze, is fixed to the inside surface and capillary forces return the condensate to the evaporator.

In the heat pipe the evaporator position is not restricted and it may be used in any orientation. If the heat pipe evaporator is placed in the lowest position, gravitational forces will assist the capillary forces.

The main regions of the heat pipe are shown in Fig. 10.

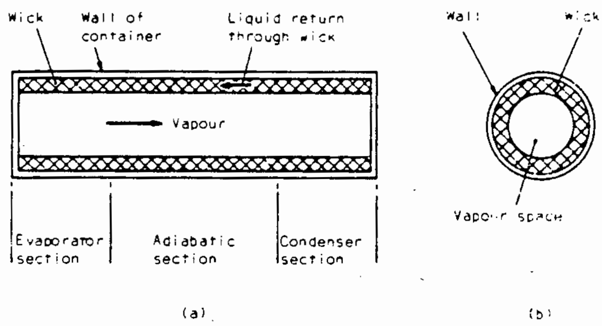


Fig. 10 The main regions of the heat pipe

In the longitudinal direction, Fig. 10(a), the heat pipe is made up of an evaporator section and a condenser section. The cross-section of the heat pipe, Fig. 10(b), consists of the container wall, the wick structure and the vapour space.

A tubular heat pipe of the type illustrated in Fig. 10, using water as the working fluid and operated at 150°C would have a thermal conductivity several hundred times that of copper.

The power handling capability of a heat pipe can be very high; pipes using lithium as a working fluid at a temperature of 1500°C will carry an axial flux of 10-20 kw/cm.

For many applications the cylindrical geometry heat pipe is suitable but, heat pipes can be made in several geometries; flat plate, flexible and annular versions are all possible. Gravitational forces will assist the capillary forces if the heat pipe evaporator is placed in the lowest position.

To illustrate the application flexibility of heat pipes the following methods of condensate return are used:-

GRAVITY	THERMAL SYPHON
CAPILLARY FORCE	STANDARD HEAT PIPE
CENTRIPETAL FORCE	ROTATING HEAT PIPE
OSMOTIC FORCES	OSMOTIC HEAT PIPE

Dunn and Reay[1] also conclude that the heat pipe is characterised by:-

(i) very high effective thermal conductance;

(ii) the ability to act as a thermal flux transformer, as illustrated in Fig. 11;

(iii) an isothermal surface of low thermal impedance.

The condenser surface of a heat pipe will tend to operate at uniform temperature. If a local heat load is applied, more vapour will condense at this point, tending to maintain the temperature at the original level.

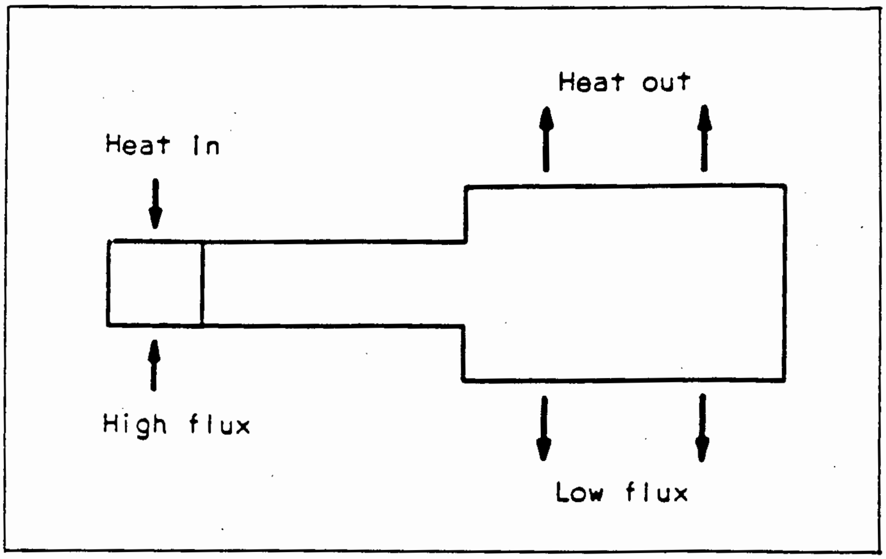


Fig. 11 The heat pipe as a thermal flux transformer

These characteristics are of special significance to this project and in later chapters of this paper the ability of the heat pipe to modulate temperature influx in an internal combustion engine application will be discussed. Modulation of engine temperature contributing to the control of detonation and emissions.

2.10 VARIABLE CONDUCTANCE HEAT PIPE

The variable conductance heat pipe (VCHP) is identified by Dunn and Reay[1] and its ability to achieve a variable thermal impedance, a feature of research interest for this project. This characteristic suggested that a heat pipe suitably modified could provide a means of temperature control by modulating the vapour pressure within the pipe.

Research conducted by others, developed a form of the heat pipe, known as the "gas buffered" heat pipe, which will maintain the heat source temperature at a virtually constant level over a wide range of input.

This can be achieved by variation of the condensing area according to the change in thermal input at the same time maintaining a constant pressure in the heat pipe.

This feature could be used conversely for the purpose of modulation of the rate of thermal influx. That is, the rate of heat removal could be controlled thereby modulating heat removal from the primary source. Such a system would provide a control of metal temperatures in an internal combustion engine application where the temperature effects of varying load need to be controlled.

To achieve variation of the condensing area gas buffering, the heat pipe is connected to a large volume reservoir. The reservoir much larger than the heat pipe, is filled with an inert gas, arranged to have a pressure corresponding to the saturation vapour pressure of the fluid in the heat pipe. Under normal conditions, the gas-vapour interface will be situated at some point along the condenser surface as the heat pipe vapour will tend to pump the inert gas back into the reservoir.

The operation of the gas buffer is as follows:-

If the heat pipe is operating under steady conditions and the heat input is increased by a small amount, the saturation vapour temperature will increase as well as the vapour pressure. Vapour presure increases very rapidly for very small increases in temperature. The small increase in vapour pressure will cause the inert gas interface to recede, exposing more condensing surface. As the reservoir volume is large compared to the heat pipe volume, a small pressure change will give a significant movement of the gas interface. Research results show that considerable heat flux changes can be accommodated by gas buffering.

Dunn and Reay[1] emphasise that the temperature which is controlled in simple gas buffered and other heat pipes, is that of the vapour in the pipe. Also normal thermal drops will occur when heat passes through the wall of the evaporating and condensing surfaces.

The various features of the heat pipe have been reviewed in detail. Also the applications and potential refinements for automotive use have been considered as alternatives for this project.

Further theoretical aspects, however, must be reviewed to provide a basis for examining the experimental research aspects which are to follow in subsequent chapters of this thesis. From this the experimental results can be analysed and the mechanism of the heat pipe understood.

2.11 HEAT TRANSFER AND TEMPERATURE DIFFERENCES IN HEAT PIPES

Dunn and Reay[1] puts forward the theory that:-

(1) The temperature drops in a heat pipe can be represented by thermal resistances and equate to a circuit as shown in Fig. 12.

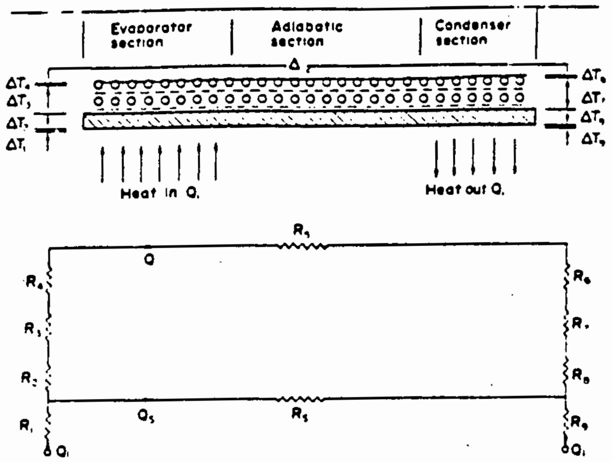


Fig. 12 Temperature drops and equivalent thermal resistances in a heat pipe

(2) Heat can both enter and leave the heat pipe by conduction from a heat source, a heat sink, by convection, or by thermal radiation.

(3) Heat by eddy currents or electron bombardment, and cooled by electron emission.

(4) Further temperature drops will occur by thermal conduction through the heat pipe walls at both the evaporator and condenser regions.

(5) Thermal resistance exists at the two vapour liquid surfaces and in the vapour column.

The processes of evaporation and condensation are now examined to identify the maximum heat transfer limits in these regions.

2.12 EVAPORATOR - HEAT TRANSFER

With low values of heat flux, heat is transported to the liquid surface partly by conduction through the wick and liquid and partly by convection, evaporation taking place at the liquid surface. By increasiong the heat flux, the liquid in contact with the wall becomes progressively superheated and bubbles will form at nucleation sites. The effect is that the bubbles will convey energy to the surface by latent heat of vaporisation and greatly increase the convective heat transfer. A critical value will be reached if the heat influx is further increased, causing burn out, at which the wick will dry out and the heat pipe will cease to function.

Earlier in this literature survey the effect of blanketing was reviewed in the context of nucleate boiling. In the theoretical and experimental chapters of this report we will be examining various considerations for both the wick-equipped heat pipe and the thermosyphon. For the latter, it is necessary to review data on heat transfer from plane, unwicked surfaces.

2.13 HEAT TRANSFER FROM PLANE SURFACES

Consider a plane heater of surface temperatures T_w immersed in a liquid maintained at temperature T_s, the boiling point corresponding to the pressure of the system. Fig. 13 shows the curve first obtained by NUKIYAMA for water according to Dunn and Reay(1) and indicates how the heat flux q will vary with T_w-T_s as the heater power is raised.

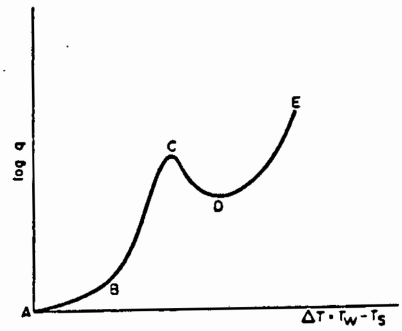


Fig. 13 Pool Boiling Regimes

Natural convection to the evaporating surface is represented by the A-B region. The region B-C is known as nucleate boiling in which bubbles form in the liquid as the heat flux is increased, achieving very high heat transfer rates with small temperature differences.

The bubbles transferring heat by latent heat and also increasing convective heat transport.

In this application as in the previous discussion, on nucleate boiling theory, the heat flux in nucleate boiling cannot be increased indefinitely, at point C a continuous vapour film forms as a result of the bubble population becoming so high that it becomes dificult for the liquid to reach the heater surface.

Under these conditions the temperature difference escalates extremely rapidly and the condition is known as blanketing, burn out, boiling crisis or critical heat flux.

Boiling is unstable in the C-D region as the surface is alternately covered by vapour and liquid. Stability returns in the D-E region, the vapour film becomes stable and the condition is known as stable film boiling. The melting point of the heater material determines point E.

The unstable boiling condition identified in the CD region will be discussed in later chapters of this thesis where the phenomena was observed during the experimental work on this project.

2.14 CONDENSER - HEAT TRANSFER

Vapour will condense on the liquid surface in the condenser. The mechanism is described as follows:-

(1) A continuous flux of molecules leaves the liquid surface by evaporation.

(2) When the liquid is in equilibrium with the vapour above its surface an equal flux of molecules will return to the liquid from the vapour and there will be no nett loss or gain of mass.

(3) If a surface is losing mass by evaporation, the vapour pressure and hence temperature of the vapour above the surface must be less than the equilibrium value.

(4) In the same way for nett condensation to take place the vapour pressure and temperature must be higher than the equilibrium value.

Based on this theory vapour condensing on the liquid surface of a condenser will experience a small temperature drop and hence thermal resistance. Further temperature drops will occur in the liquid film and in the saturated wick and in the heat pipe envelope.

Condensation can occur in two forms, either by the condensing vapour forming a continuous liquid interface or by forming a large number of drops. Generally film condensation occurs in heat pipes and is seriously affected by the presence of non-condensable gas. Vapour pumping, however, will cause such gas to be concentrated at the end of the condenser. This effectively shuts off the end of the condenser and is the basis of the gas buffered heat pipe described earlier in this report, The temperature drop through the saturated wick may be treated in the same manner at the evaporator.

A further theoretical consideration for the heat pipe, considered important in the context of this project is the interpretation of film condensation and the Nusselt theory.

2.15 NUSSELT THEORY - FILM CONDENSATION

Nusselt was the first to analyse film condensation. The theory considers condensation onto a vertical surface and the resulting condensed liquid film flows down the surface under the action of gravity and is assumed to be laminar. Viscous shear between the vapour and liquid phases is neglected. The mass flow increases with distance from the top and the flow profile is shown in Fig. 14.

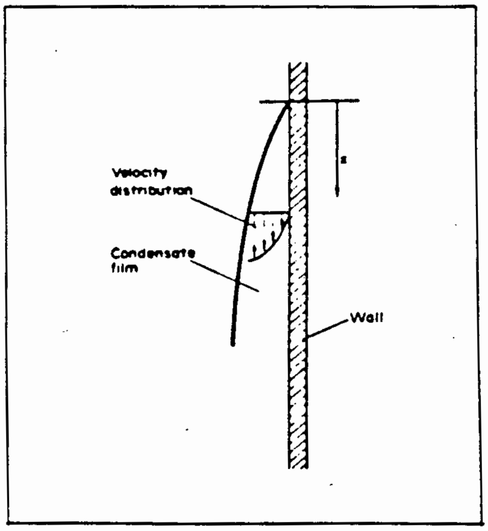


Fig. 14 Film condensation on a vertical surface

The average heat transfer co-efficient **h** over a distance x from the top is given by:

$$h = 0,943 \left[\frac{L\rho_{\ell}^{2}\, g k_{\ell}}{X\mu_{\ell}(T_s - T_w)} \right]^{\frac{1}{4}}$$

<u>where</u> $T_s - T_w$ is the difference in temperature across the film.

The extensive research done provides further valuable insight into the development of the heat pipe as a potential tool for internal combustion engine heat loss control. Analysis of the gravity assisted concept is of particular interest to this project in view of the lack of data available for either the heat pipe or thermosyphon for internal combustion engine applications.

Thermosyphon relying on gravity return of condensate, whereas the gravity assisted heat pipe performance may be improved with this feature. Both concepts were examined in this project for the purpose of experimental development and are dealt with in the relevant sections later in this report. A discussion on gravity assistance in this section is therefore relevant.

2.16 GRAVITY ASSISTED HEAT PIPES

Recent developments indicate that the gravity assisted heat pipe as opposed to reliance on the simple thermal syphon, has only received detailed attention in the last decade. The main areas of study being centred on the need to optimise the fluid inventory.

Amongst others, work was carried out by Abwat and Nguyenchi[13] at IKE Stuttgart on gravity assisted copper/water heat pipes, retaining a simple mesh wick located against the heat pipe wall. The basis of their research compared their experimental results with the theory model proposed by Kaser[15]. Tests were carried out with heat pipes at a number of angles to the horizontal, retaining gravity assistance with fluid inventories of up to 5 times that required to saturate the wick. The results indicated limitations due to nucleate boiling and were inconclusive.

Strel'tsov 1975[14] contributed to the performance of gravity assisted heat pipes and thermal syphon. He carried out a theoretical and experimental study to determine the optimum quantity of fluid to use in gravity assisted units. He derived expressions to determine the fluid inventory without quantifying the heat fluxes involved.

In view of the experimental procedure adopted for this project and the results obtained, of particular interest was his observation that film evaporation, rather than nucleation, occurred under all conditions in the evaporator section, using water and several organic fluids.

Strel'tsov[14] derived the following expression for the optimum fluid inventory for a vertical heat pipe:

$$G = (0{,}8\ell_c\ell_a + 0{,}8\ell_e)\sqrt[3]{\frac{3Q\mu_1\rho_1\pi^2D^2}{Lg}} \quad \text{---2.1}$$

where Q = the heat transport (WATTS)

and G = fluid inventory cm^3

For a given heat pipe design and assumed temperature level the dependence of the optimum quantity of the working fluid is given by the expression:-

$$G = K^3 \surd Q \quad \text{-----} \ 2.2$$

where K is a function of the particular heat pipe under consideration.

Strel'tsov - predicted the performance of a heat pipe, equation 2.1, using methanol as the working fluid at a vapour temperature of 55°C and compared this with measured values in Fig.15.

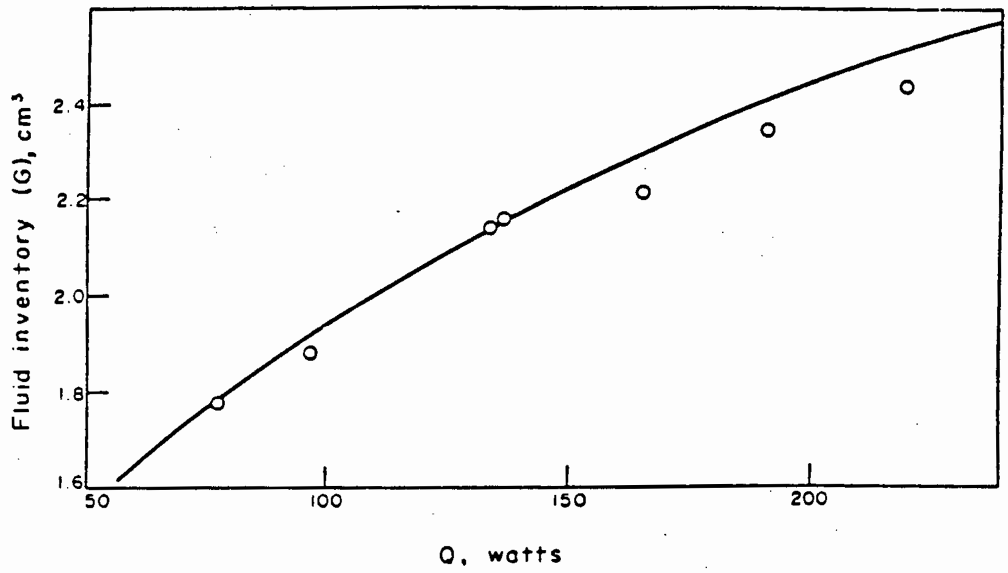


Fig.15 The measured effect of fluid inventory on the performance of a methanol heat pipe, and a comparison with results predicted by equation 2.1

The importance of working fluid inventory is stressed when considering small heat pipes.

It is apparently common practice to use a slight excess of fluid above that required to saturate the wick. However, if the vapour space has a small volume a noticeable temperature gradient can exist at the condenser, similar to that indicating the presence of non-condensable gas. This can impair the heat pipe performance as a result of the reduction of the effective length of the condenser.

On the other hand if the fluid inventory is deficient, the heat pipe may fail due to the inability of the artery to fill.

Further work was carried out by Vasiliev and Kiselyov on analytical models of vertical and arterial heat pipes. These researchers used arteries in conjunction with grooved evaporator and condenser surfaces. The artery is used to transfer condensate between the condenser and evaporator, thus providing an entrainment-free path, as distinct from a capillary type wick, while triangular grooves in the evaporator and condenser wall are used for distribution and collection of the working fluid. It is claimed that this design has a higher effective thermal conductivity than a simple thermal syphon, particularly at high heat fluxes.

THE FOREGOING SUMMARISES THE RESEARCH WORK BY DUNN AND REAY ET AL. IT IS NOT INTENDED IN THIS CHAPTER TO FULLY ANALYSE THE TOTAL THEORETICAL AND RESEARCH DATA AVAILABLE ON THE HEAT PIPE PER SE. RATHER TO EXPOSE ITS FEATURES AND TO SHOW CREDIBILITY FOR ITS CONSIDERATION IN A POSSIBLE NEW FIELD, I.E. AUTOMOTIVE, INTERNAL COMBUSTION ENGINE HEAT LOSS AND COOLING MANAGEMENT.

The basic elements of the construction and supporting theoretical principles, highlighted and discussed, are considered adequate for the researcher to understand the heat pipe and its construction. The compiling of the data into this one thesis will assist future students of the heat pipe to obtain a quick understanding of its fundamental principles.

It was found that a minimum of data existed for the application of heat pipes to internal combustion engines and that little work had been performed by researchers to gain a complete understanding of its effect on automotive application.

This literature survey therefore serves the dual purpose of conducting the initial research for the design of heat pipes and its application, as well as investigation into internal combustion engine cooling and heat losses.

The work to be done on this concept for this project can therefore be regarded as original or relatively new research.

The test results will provide a data base for any future research on automotive heat pipe applications should the project show potential for continuing this avenue of research.

2.17 CONVENTIONAL COOLING

Some research on a conventional automotive cooling system was necessary for this project in order to establish base line data for heat flow.

The reduction of lead levels in petrol combined with the trend towards high compression ratio operation and the increasing use of turbocharging is aggravating the problem of detonation in automotive spark ignition engines. Resistance to detonation can be improved by increasing the EFFECTIVENESS of the cylinder head cooling system.

According to Finlay and Parks(5), the design of an effective cooling system requires knowledge of local coolant temperatures and velocities, surface heat fluxes and heat transfer coefficients.

At present this information is not available because of the complex geometry of the coolant flow passages in most engines which makes it difficult to determine accurately local liquid velocities. A case for further precision cooling investigation. Also as shown later in this report, nucleate boiling phenomenon is present in local regions of the cylinder head.

Nucleate boiling complicates the mechanisms of heat transfer between surface and coolant and to date its influence on heat transfer in liquid cooled cylinder heads has not been satisfactorily described.

In view of the work done by Southampton University and investigations by Finlay and Parks[5], it was decided to concentrate on and establish the heat losses, heat transfer and cooling effects of various experimental concepts on an automotive cylinder head.

This project, therefore, reports on the influences of precision cooling, the heatpipe/thermal syphon and conventional cooling methods on a typical cylinder head. For each of these, the parameters and results obtained are compared and discussed in subsequent chapters.

This literature analysis has covered various aspects of heat losses in internal combustion engines and several concepts have been observed and discussed.

The following chapter will extend the theories discussed in this section and will blend the facts to form a comprehensive connection of the various cooling concepts to heat losses in internal combustion engines.

CHAPTER THREE

THEORETICAL DEVELOPMENT

The literature survey chapter has covered various aspects of internal combustion engine cooling and heat losses. Several concepts were observed, analysed, discussed and subsequently the facts were collected.

The aim of most of the papers and research data reviewed, was to investigate the effects of the various theories put forward on the cooling performance of internal combustion engines and provide direct information on the complementary aspect of heat losses.

A departure from the conventional was the introduction of the heat pipe concept and its potential for controlling cooling and heat losses in internal combustion engines.

Much common ground was covered in the examination of the various concepts reviewed, for example nucleate boiling theory, Nussels equation and heat transfer coefficients.

This chapter will draw from the discussion in the literature analysis and combine the facts to gain a complete understanding of the capabilities of the various heating/cooling concepts dealt with in this paper.

The theoretical development of the following concepts are reviewed:

3.1 HEAT PIPE/THERMO SYPHON CONCEPT.
3.2 CONVENTIONAL COOLING.
3.3 PRECISION COOLING AND HEAT LOSS CONCEPT.

3.1 THE HEAT PIPE/THERMOSYPHON

3.1.1 Design and Construction Considerations

This section will illustrate the theoretical development of a typical heat pipe. A proposed design is reviewed and calculations are performed based on the assumptions identified.

The design power of the proposed heat pipe was chosen to suit a particular internal combustion engine and calculations were done in accordance with the methods of S W CHI[2].

The three basic components of a heat pipe are:-

(i) the working fluid;
(ii) the wick or capillary structure;
(iii) the container.

In the selection of a suitable combination of the above, a number of conflicting factors may arise and the power of a heat pipe may be limited by any of the following factors, D Naeser[17]:-

(1) The heat pickup ability of the evaporator end of the heat pipe depends on the area of the evaporator surface.

(2) The capillary pressure of the wick. This is the pumping pressure within the heat pipe which has to overcome the flow resistance of both liquid and vapour within the wick structure. The maximum capillary pressure is dependent on the surface tension of the liquid and the pore size of the wick.

(3) The internal flow resistance of the liquid returning from the condenser to the evaporator within the wick structure.

(4) The vapour pressure drop due to the vapour flow within the vapour space from the evaporator to the condenser section.

(5) The maximum temperature drop across the heat pipe container tube, liquid and the structure to the liquid /vapour interface may limit the maximum heat transport.

(6) If the heat flux from the evaporator container through the liquid and wick is too high, boiling of the heat pipe fluid may take place within the wick. This will lead to drying out of the wick and localised overheating.

(7) Liquid/vapour entrainment which is the picking up of the returning liquid from the wick surface by the vapour flowing in the opposite direction may result in a reduced liquid flow to the evaporator and hence reduced performance.

3.1.2 Heat Pipe Design: Criteria/Assumptions

For the purpose of designing a suitable heat pipe, it is assumed that a 10 kw petrol single cylinder engine will need to be cooled. For example a stationary power unit envisaged for rural power application.

It was shown in the literature analysis that heat loss to water as a percentage of energy input is twenty two percent (22%).

In the 10 kw power unit this will be equal to 2 kw to be dissipated by the proposed heat pipe.

2 kw, therefore, was chosen as the design power of the heat pipe.

The design and calculations followed the methods developed by S W CHI[2].

NOTE: Most of CHI's charts were presented in imperial values, but the extracted data was converted to metric units for the calculated values in this chapter.

An arbitrary size of heat pipe was selected based on physical limitations and practicality of installation in an internal combustion engine of this size. The heat pipe was assumed to be 12 mm outside diameter and 0,2 metres long (half evaporator and half condenser).

The 12 mm O.D. pipe to transfer 2000 watts of heat at 125°C with a wick of wrapped screen type being the most common type utilised.

3.1.3 Fluid and Materials Selection

Having arbitrarily selected the basic parameters for the heat pipe, the design is initiated by selecting a fluid and material.

For a pipe operating at 125°C, water or methanol is a suitable working fluid (refer Appendix 3.4 Fig.6-1). Reference to Appendices 3.4/3.5, Fig. 6-2 and 6-3 indicates, however, that water has better liquid transport and conductance properties than methanol.

Therefore water is chosen as the working fluid.

Although Fig. 6-1 (Appendix 3.4) shows that water is compatible with copper, nickel and titanium materials, Fig. 6-7 shows that copper has superior conductance characteristics at 125^{o}C. Copper also has a cost advantage.

A copper/water heat pipe arrangement was selected.

The size of the pipe diameter necessary for a given application should be determined so that vapour velocity is not excessive. Control of vapour velocity is required since at high Mach numbers the flow compressibility of vapour contributes to a large axial temperature gradient. For a maximum Mach number of 0.2 the vapour can be considered incompressible and the axial temperature gradient negligently small.

In examining the axial heat flux, Fig. 7-1C (Appendix 3.6) indicates that at Mach No. equal to 0.2 the flux for a 12 mm diameter pipe is of the order of $10^{5.1}$ BTU/HR. = 29.3 kw. Compressibility therefore is not a problem for the selected design of 2 kw capacity.

3.1.4 <u>Container Tube Dimensions</u>

The container tube dimensions can now be determined. Fig. 7-2 (Appendix 3.6) indicates that at 125^{o}C the water vapour pressure is 100 lb/in^{2} = 68947 N/m^{2} = 689,47 kpa and copper UTS is 20 kpsi (Fig. B-3 Appendix 3.11).

Fig 7-3 (Appendix 3.8) shows that the required diameter ratio outside to inside do/di = 1.05. Table 7-1 (Appendix 3.7) however, indicates that a 12 mm O.D. pipe with a wall thickness of 0,7 mm (22 W.6) has an O.D./I.D. ratio of 1,126 and an I.D. equal to 11,28 mm.

The tube chosen, therefore, as the heat pipe container, has a 12 mm O.D. and 11,28 mm I.D.

3.1.5 End Cap Thickness

With the water vapour pressure Pv equal to 689,47 Kpa and the copper container material UTS = 20 kpsi (Fig. B-3 Appendix 3.11) the end cap thickness to diameter ratio is equal to 0,075 (Fig. 7-4, Appendix 3.8).

Therefore the required end cap thickness = 0.075 x 12 mm
= 0.95 mm

3.1.6 Wick Design

The wick design for a heat pipe is critical and S W CHI[2] adapts an elaborate method for calculating the optimum size. The calculations for this particular heat pipe wick are therefore detailed in Appendix 3.3.

The salient features of the wick design as concluded from the calculations are as follows:-

- The heat pipe condenser elevation is assumed at 25 mm above the evaporator, resulting in a hydrostatic height of 36,10 mm.
- The hydrostatic pressure = 0,374 kpa.
- With a maximum capillary pressure of 0,75 kpa the wick wire screen has a 100 mesh.
- The wick cross-sectional area = $28,765 \times 10^{-6}$ m^2.

3.1.7 Designed Heat Pipe Characteristics

In summary, a heat pipe has been designed with the following specifications:-

WORK FLUID	WATER
CONTAINER MATERIAL	COPPER
WICK MATERIAL	COPPER
CONTAINER O.D.	12 mm
CONTAINER I.D.	11,28 mm
VAPOUR CORE DIAMETER	9,52 mm
END CAP THICKNESS	0,95 mm
WIRE SCREEN MESH Number	100

Substituting these values into the equations S W CHI[2] and as scheduled in Appendix 3.3, it is shown that:-

Maximum capillary heat transport factor for the designed pipe:

$$(QL)_c max = \frac{Pcm - \Delta P_{\perp} - \rho_1 gLtSIN\Psi}{F_1 + F_v}$$

$$= 138,5 \text{ Watt-metres}$$

Whereas the capillary heat transport factor required is:-

$(QL)_c = d_v(L_e+L_c)Q$

= vapour diameter (length evaporator + length condenser) x heat flow required

= 9,52 mm (0,1 m + 0,1 m) x 2000 watts

= 3810 W-m.

3.1.8 Initial Design Conclusion and Performance Capability

A heat pipe 12 mm diameter and 20 cm long with a capillary wick is inadequate to conduct a heat flow of 2 kw.

The heat pipe was an arbitrary choice based on the physical space limitations in the confined space of a typical small power unit.

The designed pipe heat transport performance would have to be increased by a factor of 20 times to approach the heat loss required.

Earlier in this chapter, reference was made to a large number of conflicting factors which may arise to limit the power of the heat pipe. The combination of those factors, and the use of empirical tables, charts and data placed a reservation on the calculated results of the design.

The negative results obtained with the theoretical performance of the proposed heat pipe and the identified conflicting factors (NAESER[17]), prompted an additional literature search to obtain verification of the final design results.

Dunn and Reay[1] was consulted for details of a typical copper/water heat pipe evaluation and its potential performance. This source manufactured and tested a copper heat pipe using water as the working fluid, to determine the temperature profiles and its maximum capability.

The evaluation employed a copper/water pipe of similar dimensions to the heat pipe design considered for this project, i.e. a 12 mm O.D. pipe but 320 mm in length whereas 200 mm was used for the calculated heat pipe design.

The Dunn and Reay[1] heat pipe operated vertically with gravity assistance. The power was applied and on achievement of a steady state condition, the thermocouple readings and temperature rise through the water jacket were noted, as was the flow rate.

The test results are shown in Fig.1.

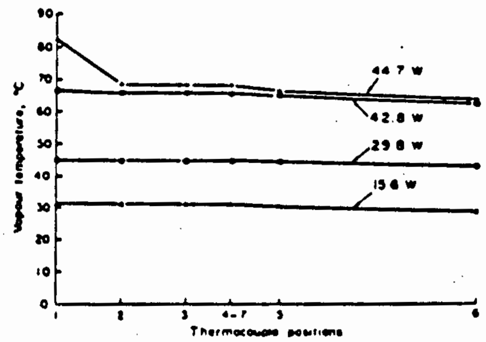


Fig.1 Typical temperature profiles along a heat pipe under test

These indicate that:-

The best result at vapour temperature of 85°C = <u>44,7</u> watts.
The <u>designed</u> <u>pipe</u> water temperature of 120°C = 138 W-m
= <u>28,0</u> watts.

In another example Dunn and Reay[1] tabled the power capabilities for a 9,5 mm O.D. copper heat pipe of length 30 cm and water as the working fluid. The evaporator above condenser by 18 cm:-

The results obtained were:-

Vapour Temperature	Power Out
84°C	17 watts
121°C	30 watts
162°C	54 watts
197°C	89 watts

Both these examples confirm that the heat pipe designed for this project produced heat transport values of the same order, therefore validating the design calculations.

The heat pipe designed within its physical size limitations i.e. 12 mm O.D. and 20 cms in length is inadequate to dissipate the heat equivalent of the 2 kw required to cope with the heat loss criteria established for the design.

From the above it is seen that a radical approach to the heat pipe concept for internal combustion engines is necessary to achieve the performance required, within the confines of the limited space available in automotive cylinder head castings.

3.1.9 Alternative Heat Pipe Configuration

The inadequate performance of the designed conventional heat pipe, led to the examination of alternative heat pipe configurations and consideration given to the following:-

(1) Simulation of the heat pipe configuration within the cylinder head itself by utilising the existing water jacket as the evaporator. A fabricated pipe and condenser unit being connected to the cylinder head to complete the heat pipe/thermosyphon arrangement. (Refer Fig.2 and Appendix 4.5 and 4.6.)

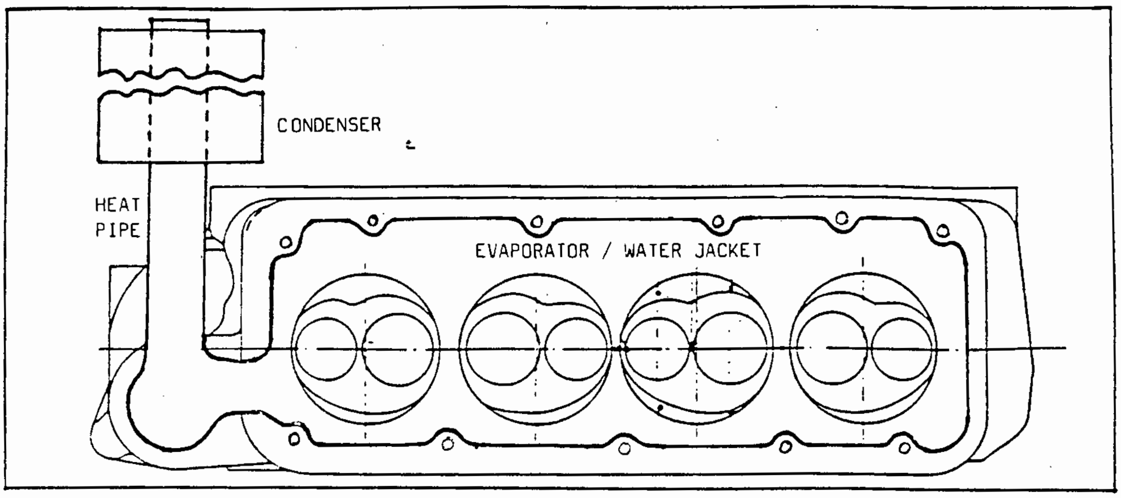


Fig.2. Integrated Cylinder Head and Heat Pipe

(2) The performance of a Gravity thermosyphon could be tested using the 12 mm O.D. container again and compared against the wick design with capillary action used in the calculated design.

(3) If a thermosyphon performance is adequate it may be worthwhile considering for use in conjunction with proposal in item (1) above.

(4) Introduce a means of modulating the heat pipe performance to cover the high temperature condition required at engine idle through to the lower temperature condition when accelerating on wide open throttle (W.O.T.). A temperature modulation module is possible by increasing the pressure within the heat pipe. This could be achieved by the gas buffer phenomena as discussed in the literature survey. The effects and results to be examined for automotive application.

The possibility of "modulating" the cooling as required, identifies the major problem of achieving a sufficiently rapid change of surface temperatures in the cylinder head. From an idle condition, full power may be in demand one second after depressing the throttle pedal fully.

Thus, after a period of idling, with say the cylinder head surface temperature well above normal, it would be necessary to reduce the surface temperature to a subnormal value within a second or two to avoid violent detonation. The flow modulated heat pipe may be the means for achieving this result and the experimental apparatus will be investigated for this possibility.

3.1.10 <u>Flow Modulated Heat Pipe</u>

It is appropriate to consider the modulated heat pipe at this stage to gain an insight into its behaviour under modulated and unmodulated conditions. The concept will have a bearing on the experimental results and subsequent discussion on the phenomena observed during the experimental research.

The concept of heat pipe performance modulation is explored by S W CHI[(2)] and refers to the "VAPOUR-FLOW MODULATED HEAT PIPE".

This is achieved by throttling the vapour flow between evaporator and condenser, creating a pressure difference between the two sections and hence a corresponding temperature difference. Temperature characteristics of heat pipes can be varied using this principle. Refer Fig. 3.

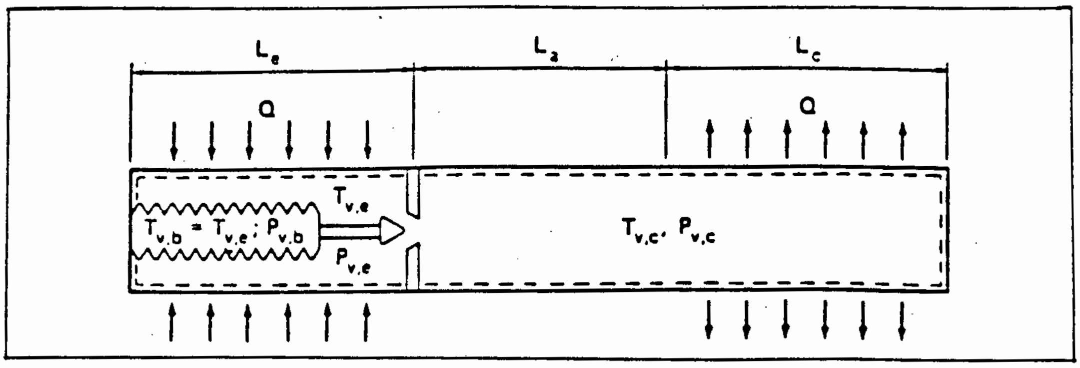


Fig. 3 Control Fluid Capsule

Variable modulation may be achieved by (Fig. 4):-

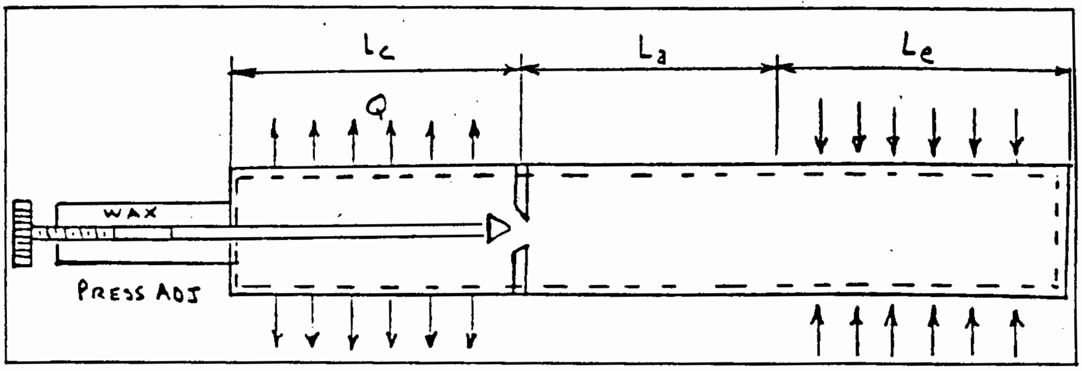


Fig. 4

Experiments on pressure modulation of a heat pipe based on engine power demand will be the subject of a separate study or could become an extension of the current study. This will depend on the final conclusion and investigation into the potential of heat pipes to satisfy internal combustion engine cooling parameters.

3.1.11 Experimental Integrated Heat Pipe Fabrication and Test

It was decided to continue with the programme of cooling by the heat pipe concept, but in a different configuration.

In this case the total cylinder head is to simulate the heat pipe configuration within the cylinder head itself as proposed in alternative (1) of section 3.1.9.

The cylinder head water passages to act as the evaporator section and the water passages leading to an external condenser. The condenser to be mounted vertically to allow the condensate to return by gravity. No wick arrangement being proposed due to the complex configuration of the cooling jackets in the cylinder head.

According to Dunn and Reay[1] cleanliness in heat pipes is a prime importance to ensure that no incompatibilities exist and to make sure that the wick and wall will be wetted by the working fluid. Above all the heat pipe must be leak tight to a very high degree.

The above requirements put a reservation on heat pipe applications for automotive cooling purposes. Especially if the total cylinder head is to be treated as a complementary part of the heat pipe. Due to impurities in the manufacture of automotive cylinder heads and water jackets and some retention of moulding sand, a wick operated integrated heat pipe would seem to be impractical. Therefore at best only a thermal syphon can be considered.

A further reservation is the strict criteria of a leak tight system. Automotive castings are prone to porosity and leak problems, therefore it would appear that even if a heat pipe concept could be made operational it is questionable whether consistency of performance could be maintained.

Despite the above reservations it was decided to evaluate the integrated heat pipe concept for automotive applications. The proposed experimental work on this new system will extend the research on "heat losses in internal combustion engines" and expands the data available for future development of the concept.

The experimental apparatus to be fabricated as a thermal syphon to verify the theoretical aspects of the concept.

The research investigation on this apparatus will be to determine the heat transfer ability in this configuration.

Heated oil (100^oC-180^oC) will be passed into the combustion chamber and through the exhaust port of the cylinder head. Heat inflow and outflow will be measured. The valve bridge and exhaust port temperature will be measured continuously during the tests. (Ref Appendices 3.1 and 3.2.)

The performance of the thermal syphon and its ability to extract heat will be measured by establishing the volume of cooling water flow through the condenser and the coolant inlet and outlet temperatures. This will be compared with the valve bridge temperature response.

Details of the apparatus are shown in the Appendices 4.5 to 4.9.

3.1.12 Oil Heating Medium (Volume/Velocity)

Assumptions:

INLET TEMPERATURE OIL = $150^{o}C$, OUTLET $140^{o}C$, T = $10^{o}C$.

Heat dissipation required = 2000 watts

= (3.4121 x 1.055 x 2000)KJ/h

= 7200 = 120 KJ/min.

OIL HEAT FLOW =

$$120 \text{ KJ/min} = \frac{\text{Sp heat} \times \text{density} \times T \times \text{flow(cc/min)}}{10^6} =$$

$$\text{FLOW} = \frac{120 \times 10^6}{2.421 \times 880 \times 10} = 5632 \text{ cc/min}$$

Therefore test rig pump to deliver:

approximately 6 litres/ minute.

Full details of the integrated heat pipe test rig will be covered in the chapters on experimental procedure and description of research apparatus.

3.1.13 Optimum Fluid Inventory

Chapter 2 provides an expression for determining the optimum fluid inventory of a heat pipe, namely:

$$G = (0,8\ell_c\ell_a + 0,8\ell_e)\sqrt[3]{\frac{3Q\mu_1\rho_1\pi^2D^2}{Lg}}$$

Where G = Fluid inventory in cc

Q = Heat transport in watts.

The remainder of the terms are defined in Appendix 3.12.

The approximate fluid inventory for the experimental heat pipe can now be established assuming the following:-

(1) equal lengths for the evaporator, condenser and adrabatic section;

(2) ± 1,000 watts heat transport (maximum capability of test rig (refer Experimental Design and Operation, chapter 6 - Heat Pipe Performance).

Substituting these values into the above equation the calculation is processed in Appendix 3.12, indicating that:-

The required fluid inventory is = 303.6 cc.

For the experimental tests 300 cc of water was adopted as the basic fluid inventory requirement and variations of this amount are covered in the experimental procedure, the results are analysed and finally discussed in chapter 6.

3.1.14 Vapour Compressibility and Flow

One of the performance limiting factors of the heat pipe suggested by NAESER[17] was vapour pressure drop. In order to interpret the results of tests in a later chapter, the basic theory of compressible flow is reviewed.

The axial mass flow in a cylindrical heat pipe increases along the length of the evaporator region to a maximum value at the end of the evaporator, and decreases in the condenser region.

The flow velocity rises to a maximum value at the end of the evaporator region where the pressure will have fallen to a minimum.

Other researchers have indicated that the flow behaviour for a heat pipe is similar to that of a gas flowing through a converging diverging nozzle. For the heat pipe the area remains constant but mass flow varies, whereas the mass flow is constant but cross-sectional area is changed for the nozzle.

Kemme has also shown that a heat pipe can operate in a very similar manner to the converging diverging nozzle. His experimental arrangement and results are shown in Fig. 5, which show the temperature profiles in a heat pipe, whilst Fig. 6 shows the pressure profiles in a converging diverging nozzle.

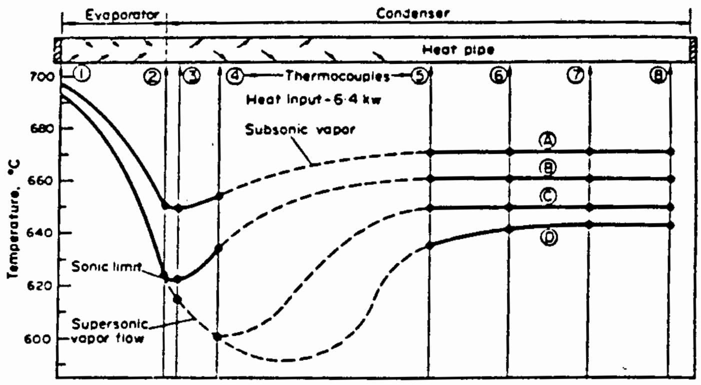


Fig. 5

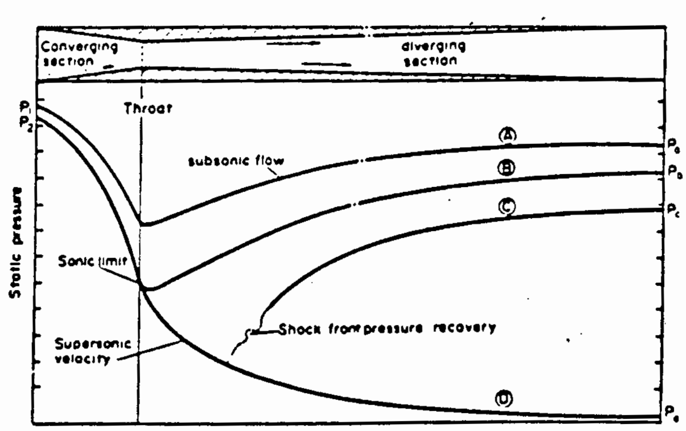


Fig. 6

Kemme maintained a constant heat input of 6.4 kw and measured the axial temperature variation. As the axial temperature variation is directly related to pressure, the temperature profile is considered to be the same as the pressure profile. The heat rejection at the condenser was varied by means of a gas gap. For Fig. 5 - Curve A shows subsonic flow with pressure recovery; Curve B, by lowering the condenser temperature achieved sonic velocity at the end of the evaporator and therefore operated under choked flow conditions. Decreasing the thermal resistance between the condenser and heat sink simply reduced the condenser region temperature but did not increase the heat flow which was limited by the choked flow condition and fixed axial temperature drop in the evaporator. Under these conditions the heat pipe operation will not be isothermal as considerable axial temperature and pressure changes will exist under sonic limitation conditions.

3.2 CONVENTIONAL COOLING

The various concepts to be researched in this project have been identified and earlier in this chapter it was confirmed that a small automotive cylinder head would form the basis of the experimental apparatus.

A typical cylinder head as used in the experimental tests is shown in Fig. 7. The exhaust ports are clearly seen as well as the thermostat aperture which served as the base for connecting the heat pipe reviewed in the previous section.

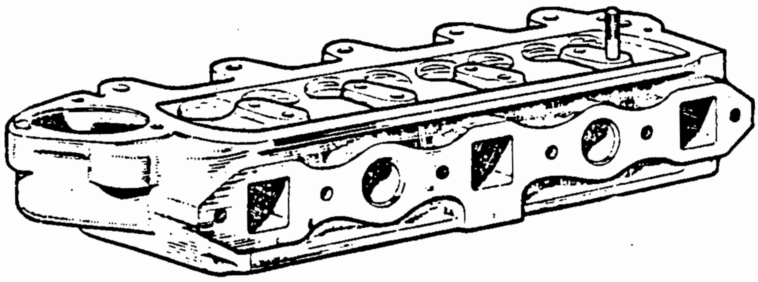


Fig. 7 Typical Automotive Cylinder Head

The rationale for this apparatus being, that simulation of actual conditions can be achieved by utilising heated oil to substitute for combustion gases, passing through the combustion chamber and out of the exhaust port. This allows temperature readings of the valve bridge and valve area to be recorded, whilst establishing coolant temperatures and heat loss levels for the different systems. The conventional cylinder head apparatus therefore results in a compact benchtop experiment without the complexity of performing the research on a complete engine.

This arrangement has had the advantage of the heat pipe and precision cooling concepts being tested under directly applicable conditions. In addition tests can be conducted with the conventional cooling system, again under simulated actual conditions, thereby establishing base-line data. This data forming the basis for a direct comparison of performance with the test results obtained from the two concepts.

For the conventional cooling tests, coolant water will be allowed to flow through the cylinder head in the normal manner from a mains supply under controlled pressure and flow. The criteria would be to hold the valve seat and valve bridge temperatures at similar values, to the other two system tests, i.e. approximately 130^{o}C, as the apparatus bulk oil temperature is predicted to be 175^{o}C. The heat outflow of the coolant water to be monitored.

A theoretical prediction of an effective cooling system, however, requires knowledge of the local coolant temperatures and velocities, surface heat fluxes and heat transfer coefficients. Because of the complex geometry of the coolant flow passages in a conventional cylinder head it is difficult to determine accurately the local liquid velocities. Nucleate boiling could be present in local regions of the cylinder head and complicates the mechanisms of heat transfer between surface and coolant.

For the conventional cooling concept, a heat transfer model is predicted, incorporating forced convection and possible nucleate boiling. The results to be compared with the measured values obtained experimentally.

3.2.1 Apparatus Heat Input

The philosophy of using heated oil to simulate combustion gas temperature is explained above. Using the following assumptions, the heat input of the heating oil and the heat outflow of the coolant, the heat transfer values of the conventional concept can be predicted.

Projected bulk oil temperature	= 175^{o}C
Oil Flow at 175^{o}C	= 12 litres/min

<u>Assumed temperatures</u>

Valve temperature = 130°C

Oil temperature in = 168°C

Oil temperature out = 166°C

Oil characteristics - refer Appendix 4.14

Oil heat in = Sp HT x Density x Flow x Temp Difference

= 2.42 x 880 x 12000 x 2

= 51,11 KJ/min

= 852 watts

Now in the literature survey chapter 2, Holmes[4] has shown under 2.1 "Efficiency and Heat Balance" that heat losses within a standard engine are generally equally distributed between the pistons, cylinder head and cylinder block, as shown in Fig. 8.

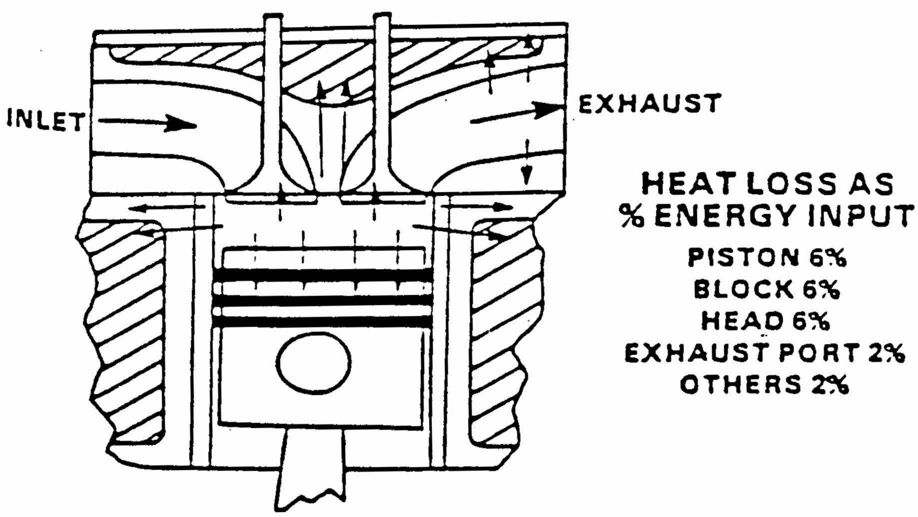


Fig. 8 Heat Flow in a Standard Engine

Also that a typical energy balance is as Fig. 9.

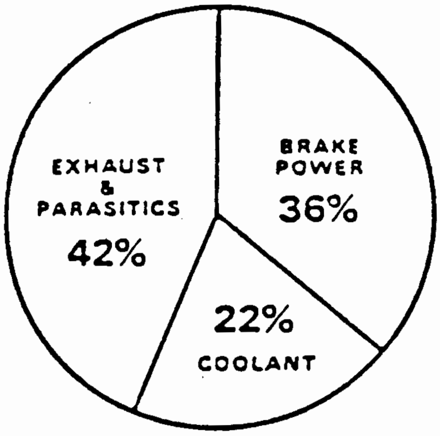


Fig. 9 Typical Energy Balance

If we now assume that as the experimental apparatus consists of a cylinder head only, no heat loss to brake power as a percentage energy input occurs. Then 100% of the **heated oil input** viz 51.11 KJ/min has to be balanced or dissipated in the cylinder head and coolant.

3.2.2 Conventional System Predicted Performance

Using the typical energy balance we can therefore distribute the total heat input between the exhaust, parasitics and coolant.

Total heat input = 51.11 KJ/min

The ratio of coolant to exhaust = 22:42 = 52%

$\therefore$ Coolant heat outflow is expected to be 0.52 x 51.11

= 26,5 KJ/min.

= 442 watts

For our conventional concept the following performance is predicted:

Heat input = 51.11 KJ/min = 852 watts
Heat outflow = 26.5 KJ/min = 442 watts
Ratio Heat out/in = 52%

3.2.3 Predicted Coolant Flow

If a temperature differential for coolant water in and out is assumed at 55°C, then water flow is predicted at:-

Coolant heat = 26.5 KJ/min (refer 3.2.2)
Water heat flow = Sp heat x density x Temp Difference x flow

$$\text{Coolant Flow} = \frac{26.5 \times 10^6}{4.186 \times 998.6 \times 55} = 115.2 \text{ cc/min}$$

These values, therefore, indicate the predicted performance based on the anticipated test rig achievable temperature and calculated value for the oil heat input.

It is expected to achieve heat transfer value of the same order under actual experimental conditions.

3.3 PRECISION COOLING

In chapter 2 the research conducted on precision cooling was summarised and the salient points relating to this project identified. It was shown that heat transfer from the walls of the cylinder head to the coolant assumed to take place by a combination of forced convection and subcooled nucleate boiling.

Using the equations developed by this authority and an arbitrary layout for a small bore test apparatus, a theoretical performance model was calculated for a typical precision cooling configuration.

In chapter 2 it was emphasised that the test apparatus would be developed to provide a common base for testing the alternative concepts from which the experimental results will be compared to establish relative performance.

The basis for the test rig to be a small automotive cylinder head and all temperature measurements to be made on one cylinder of the head. Construction of the test apparatus will be detailed in chapter 4.

3.3.1 Theoretical Performance

The theoretical calculations are based on the principle of small bore coolant tubes incorporated into the cylinder head deck above the combustion chamber as described in section 4.10, chapter 4.

Two 6 mm I.D. tubes were fitted and their heat transfer ability was calculated according to equations proposed by Mansfield and Grover [11].

$$\text{Viz:-} \quad \frac{hd}{k} = 0.023 \left(\frac{Pvd}{\mu}\right)^{0.8} \times \left(\frac{c\mu}{k}\right)^{0.4} \quad \ldots \; 3.3.A$$

To arrive at values for the various symbols the following assumptions are made:-

(i) Tube size 6 mm I.D.
(ii) Water coolant
(iii) Coolant inlet temperature 20°C
(iv) Coolant flow 200 cc/min
(v) Water properties Dunn and Reay[1] page 299.

3.3.2 Coolant Velocity

Tube diameter = 6 mm = 0.006 m

$$\text{Tube area} = \frac{\Pi D^2}{4} = \frac{\Pi}{4}(0.006)^2 = 2.82 \times 10^{-5}\ m^2$$

Flow = 200 cc/min = 100 cc/min per tube

$$\text{Velocity} = \frac{100 \times 10^{-6}}{60} \times \frac{1}{2.82 \times 10^{-5}} = 5.91\ \text{cm/sec}$$

Substituting velocity and values obtained from Dunn and Reay[1] tables the equation values become:-

h = coefficient of heat transfer	w/sq cm°C
d = passage diameter	0.006 m
k = thermal conductivity of liquid	0.649 w/m°C
P = density of liquid	983.2 kg/m³
v = velocity of liquid	5.91 m/sec
μ = viscosity of liquid	0.47 kg/sec/m
c = specific heat of liquid	4.179 KJ/kg°C

Substituting into equation 3.3.a:-

$$h = 108.1 \times 0.023\ (741.79)^{0.8} \times (3.02)^{0.4}$$

Heat transfer coefficient = h = 766 $J/m^2/°c/sec$

= 766 $watts/m^{2}°c$

(at 5.91 cm/sec velocity)

also h = Constant x $V^{0.8}$
if liquid properties remain the same and one tube size is used.

In this case:- $h = 766$ watts/m²/°C = constant x $V^{0.8}$
therefore constant = $\frac{766 \text{ watts/m}^2/^\circ\text{C}}{V^{0.8}}$

Now $V^{0.8} = 0.1041$ (5.91 cm/sec)

Therefore constant = $\frac{766}{0.1041}$

$= \underline{\underline{7354}}$ (Calculation Base for Appen 6.5)

Total heat transfer for two tubes in cyl head $= 2 \times 766$ w/m²cm°C

$= 1532$ w/m²cm°C

3.3.3 Test for Nucleate Boiling

The theory of nucleate boiling has been covered in the literature survey. The precision cooling concept presents an opportunity to observe the effect of nucleate boiling.

With the precision cooling apparatus operating under stable conditions it should be possible to explore the existence of this phenomena. This can be done by raising the apparatus input temperature or reducing coolant flow. The latter having the additional effect of reducing the passage pressure which by lowering the water boiling point, should increase the boiling potential.

An important observation will be whether by raising the passage surface temperatures there will be any indication of an increase in heat transfer coefficient or whether this remains constant.

The theoretical predictions developed in this chapter are based on the previous research reviewed in chapter 2. Most of the aspects and theories associated with heat transfer using water coolant have been identified and examined. Where applicable these have been developed sufficiently to establish criteria for the experimental test programme and comparison of theory against experimental results.

CHAPTER FOUR

EXPERIMENTAL DESIGN CONSTRUCTION AND OPERATION

In chapter 2 the literature was analysed and having investigated the work performed by others, theoretical concepts were prepared and developed in chapter 3.

The objective of this section is to provide an experimental process and procedure to test the validity of the proposals and theories outlined.

4.1 EXPERIMENTAL APPARATUS - GENERAL BACKGROUND

The initial proposal for this project was to concentrate on expanding the precision cooling work done by Mansfield and Grover(11) and to construct a test rig similar to the one used at ISVR for this work. Some practical improvements for the local version were envisaged to simplify construction and provide easier testing of various small bore passage configurations.

The ISVR apparatus consisted of a copper bar 53 mm in diameter and approximately 200 mm long. A central hole was bored to accommodate an electrical heating element clamped in position. To form a test passage, a longitudinal groove was cut on one side of the copper bar and covered with a Pyrex observation window to form a semicircular 4,5 mm diameter passage.

Past experience with similar experiments, had shown that despite modulation of the apparatus electrical heating element, the change of conditions from nucleate boiling to formation of a vapour layer and "blanketing" can occur instantaneously. This causes a rapid rise in surrounding metal temperature and "burn-out" of the heater before modulation can take effect (refer Fig. 4.1). Damage to the electrical heating equipment will occur as a result of a sudden temperature rise and the thermal inertia of the apparatus.

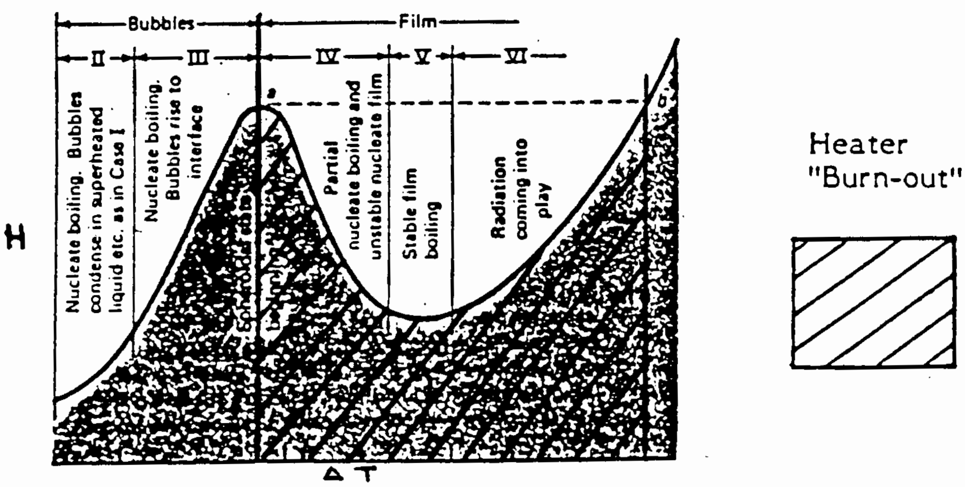


Fig. 4.1

Because of these reservations, alternative heating methods for the heat transfer apparatus were considered. Those less sensitive to heat modulation being: heated oil, steam or gas flame.

In view of the inherent problems with the electrically heated apparatus and the suspect accuracy of heat transfer performance of the apparatus proposed, it was decided to abandon this research route.

4.2 ALTERNATIVE HEAT TRANSFER CONCEPTS

In lieu of pursuing the precision small bore cooling concept only, it was decided to examine additional concepts and systems to determine the effect on internal combustion engine heat losses.

The main approach to the project was then changed from developing the precision cooling system to establishing a comparative performance investigation between this system and other selected concepts. The general parameters of the precision cooling system to be applied, viz optimum cooling of the critical components only.

It was decided to examine the relatively new heat pipe concept and to assess its performance against that of the precision cooling as well as that of a conventional system.

In chapter 3 it was shown that a heat pipe designed within the physical limitations imposed by a small internal combustion engine was, however, incapable of producing the necessary heat transfer/cooling capability required. The heat pipe transfer factor being 138,5 w/m for the designed heat pipe whereas 3810 w/m was required. It was then decided to investigate an integrated type heat pipe consisting of an automotive cylinder head as the evaporator end and a plant-on heat pipe/tube mainly comprising the condenser (refer appendix 4.6). A small automotive cylinder head was therefore chosen as the basis for the assessment of the three concepts as this would form a common base on which the various concepts to parameters could be superimposed.

4.3 APPARATUS - GENERAL ARRANGEMENT

For the heat transfer performance assessment, the apparatus features shared by the heat pipe, conventional system and precision cooling, consisted of the following:-

(i) The automotive cylinder head.
(ii) 20 litre bulk heating oil tank.
(iii) 3 kw electrical heating element for the bulk oil.
(iv) A thyristor controller and thermostat to modulate bulk oil temperature. (Appendix 4.15)
(v) Viking idler and rotary gear pump oil pump type FH32, delivery 12 litre/min. (Appendix 4.16)
(vi) 0.25 kw electric motor 2850 rpm for oil pump drive.
(vii) Oil system pressure gauge 0-100 kpa.
(viii) 12 Station switch for thermocouple selection.
(ix) Fluke digital multimeter and 80TK thermo module.
(x) Thermocouples type K to suit relevant concept.
(xi) Water system pressure gauge 0-200 kpa.
(xii) Air pressure system for heat pipe modulation.

A schematic layout of the general arrangement is shown in appendix 4.1.

The general experimental process consisted of pumping heated oil from the bulk oil tank via the oil pump into the cylinder head combustion chamber and returning through the exhaust valve port to the bulk oil tank. The oil was circulated after the bulk oil temperature stabilised at approximately 175°C

Temperature thermocouples were placed at the oil inlet to the combustion chamber and outlet oil temperature taken immediately after the exhaust port adjacent to the cylinder head. The thyristor controller modulated the bulk oil temperature to within 3°C.

4.4 HEATING OIL/HEAT INPUT

The heating oil volume flow requirement was initially calculated at 6 litres/minute based on the required heat transfer and assumed temperature differential. In chapter 3 it was shown that under "Experimental Conditions" this was found to be inadequate. The apparatus was subsequently modified by the addition of a more powerful pump motor and the bulk oil temperature was raised to 175°C. The modified system supplied approximately 12 litres/minute which was used as a basis for all the tests. The apparatus heat input was predicted in chapter 3 at 850/900 watts.

The heating oil used for the experiments was a specially selected mineral oil heat transfer fluid for indirect closed fluid heat transfer systems operating at bulk temperatures up to 320°C. The fluid known as SHELL THERMIA D has its characteristics detailed in appendix 4.14. Shell Thermia D oil has a high viscosity with good temperature/viscosity characteristics and heat transfer properties.

4.5 THERMOCOUPLE/TEMPERATURE MEASUREMENT

Metal temperatures for the cylinder head were registered by thermocouples placed in the valve bridge between the exhaust and inlet valves, also 180°C opposite in a valve seat, (refer Appendix 4.4).

Mains water, run to waste, provided the coolant for each concept test, with inlet and outlet temperatures being recorded by the thermocouples placed in the inlet/outlet tubes immediately adjacent to the cylinder head.

Additional thermocouples were required for the heat pipe and these will be reviewed later in this chapter.

All surfaces subject to possible heat loss were well lagged with fibre matting and covered with heavy asbestos rope to ensure minimum heat loss. The lagging is shown in the photographs contained in the Appendices 4.5 to 4.11.

4.6 HEAT PIPE CONCEPT

Appendices 4.5 to 4.8 show the integrated cylinder head/heat pipe arrangement and the apparatus layout.

Appendix 4.5 shows the end view of the unlagged test rig and clearly indicates the five heat pipe thermocouples inserted into the copper connectors along its length. The 2nd and 3rd thermocouple wires shown are the inlet and outlet thermocouples for the coolant water temperatures. The Fluke measuring instrument is in the foreground.

Appendix 4.6 is another unlagged view and the T-valve situated at the top of the pipe is used for evacuating, filling of the fluid inventory and also allows external air pressure to be applied for modulation experiments. In this appendix the copper pipe leading off the first "spark plug" position connects to the oil system pressure gauge and registers the pressure of the heating oil flowing through the combustion chamber.

Appendices 4.7 and 4.8 show the totally lagged apparatus.

Appendix 4.2 shows the proportion of the heat pipe/thermosyphon and its construction whilst appendix 4.9 shows the lagging applied to this component. The thermocouple connections to the pipe are clearly shown, enabling the temperature gradients to be established and recorded.

4.7 APPARATUS OPERATING PROCEDURE

The apparatus start-up procedure was the same for all three concept tests.

The bulk oil heater was switched on and the bulk oil temperature allowed to build up to the 175°C as set by the thyristor controller.

The oil pump was then started and the test rig allowed to warm up. When the bulk oil temperatures had again stabilised the experimental tests were commenced.

The heating oil flow at 175°C was established by stopwatch and container after observing three flow tests.

Prior to commencing the heat pipe tests, the heat pipe was evacuated and the fluid inventory then added. An analysis of the fluid inventory variations used for the test runs is scheduled in appendix 4.13 showing a range of fluid volumes tried experimentally. It was found that the heat transfer response was at its best using the theoretically calculated volume of 300 cc, producing repeatable results when other factors were varied. Refer chapter 3 for predicted volume and chapter 6 for discussion of results.

The procedure after commencing the test was to run the programme for approximately 180 minutes with all data readings being taken every 5 minutes. The data recorded is listed in the appendices and identified in chapter 5.

For the heat pipe nominal coolant flows were preset and the apparatus allowed to stabilise. Thereafter coolant flow, system pressure were varied and all temperatures and pressures observed. The coolant flow was also measured every 5 minutes by means of stopwatch and beakers. Metal temperature response was noted to observe the effect and response time to alteration of coolant flow and system pressure.

Several prolonged tests were done for the heat pipe experiments involving approximately 4000 temperature and data readings. Appendix 6.2 is representative of the experimental data obtained for the heat pipe. An extract from Appendix 6.2 is shown as test C results in Appendix 6.3. This Appendix also summarises the results under tests A and B for additional tests done on the heat pipe, to identify heat pipe temperature gradients.

4.8 TEMPERATURE MODULATION

The chapter on theoretical development, reviews the effects of system pressure and vapour compressibility in the heat pipe. Temperature and pressure profiles were discussed and the importance of vapour pressure to overcome sonic flow choke conditions and provide a uniform axial temperature gradient. Under these conditions the heat pipe would operate isothermally with maximum heat transfer.

This was achieved experimentally by providing an external air pressure system to increase system pressure as required. The system allowed isothermal conditions to be induced and overcome sonic choking as the vapour pressure increased with higher boiling temperature in the evaporator.

The air pressure facility was supplied from a bulk air receiver via an air pressure regulator valve providing system pressures up to 100 Kpa.

A temperature modulation system was thus provided to observe the potential for modulating the heat pipe and its future application. The results and conclusion are reviewed in later chapters.

4.9 CONVENTIONAL COOLING SYSTEM

The experimental layout and arrangement for this apparatus is shown in Appendix 4.10 for the unlagged condition and 4.11 for the lagged condition.

The two thermocouples for coolant inlet and outlet are clearly seen, whilst the remainder of the apparatus is the same as for the heat pipe system, but with the pipe and condenser removed.

The oil heating arrangement remained the same as for the heat pipe system and the heated oil being passed through the combustion chamber and exhaust at a constant rate.

The coolant water flow was connected directly to the cylinder head water jacket in the conventional manner and the flow was varied to suit the experimental data requirements.

The procedure for conducting the test was basically the same as the heat pipe except that the coolant water now absorbed heat directly from the cylinder head metal and not through a condenser. All other temperature readings, etc, being recorded as before.

4.10 PRECISION COOLING

Appendix 4.12 shows the method of incorporating the small bore 6 mm diameter coolant tubes in the combustion chamber deck and adjacent valve ports.

The cylinder head was contour machined in these areas so that continuous tubes could be laid in and welded to the cylinder head metal surface to ensure positive heat transfer.

Also shown are the thermocouple positions located in the exhaust/inlet valve bridge and surrounding valve area.

The modified cylinder head was mounted on the apparatus pedestal and heavily lagged as for the other two systems.

The oil heating procedure followed the previous arrangement with heated oil passing through the combustion chamber and through the exhaust port returning to the bulk oil tank. As before, the flow was constant at 12 litres/minute and temperature 175°C.

The experimental procedure, required the apparatus temperature to stabilise and then the coolant flow was varied to observe the effects on the cylinder head metal and coolant outlet temperatures. The coolant flow was reduced to ascertain whether nucleate boiling could be induced.

4.11 EXPERIMENTAL OBSERVATIONS/READINGS

The categories for the experimental readings were planned to be the same for the three systems and are scheduled below:-

Event time - Minutes	Valve Bridge Temp 9 - °C
Heat pipe temp 1 - °C	Valve temp 10 - °C
Heat pipe temp 2 - °C	Bulk oil temp 11 - °C
Heat pipe temp 3 - °C	Coolant outlet temp 12- °C
Heat pipe temp 4 - °C	Temp diff water - °C
Heat pipe temp 5 - °C	Temp diff oil - °C
Coolant inlet temp 6 - °C	Coolant flow cc/min
Oil heat inlet temp 7 - °C	Heat flow oil KJ/min
Oil heat outlet temp 8 - °C	Heat flow water KJ/min

CHAPTER FIVE

RESULTS

The results are presented in tabular and graphical form and are contained in the appendices listed. They have been placed at the back of this thesis so as not to interrupt the script.

5.1 HEAT PIPE RESULTS

Appendix 6.2 shows the results for the heat pipe concept tests. These results contain all the data measured, including relevant temperatures, coolant flow and heat flow results.

In the case of the heat pipe, it was essential to record the thermocouple readings of the heat pipe itself, to determine the temperature gradient variations.

Typical readings are shown below, with details scheduled in appendix 6.2. Thermocouple numbering and position are shown in appendix 4.3

Event time - minutes	Valve bridge temp 9 - °C
Heat pipe temp 1 - °C	Valve temperature 10 - °C
Heat pipe temp 2 - °C	Bulk oil temp 11 - °C
Heat pipe temp 3 - °C	Coolant outlet temp 12- °C
Heat pipe temp 4 - °C	Temp diff water °C
Heat pipe temp 5 - °C	Temp diff oil °C
Coolant inlet temp 6 - °C	Coolant flow cc/min
Oil heat inlet temp 7 - °C	Heat flow oil KJ/min
Oil heat outlet temp 8 - °C	Heat flow water KJ/min

The recorded results provide the data for calculating the heat transfer values, heat flow generated by the oil heating medium and the heat flow absorbed by the coolant water. The experimental thermocouple readings for the heat pipe provide the data to establish its characteristics under various system pressures and to determine its performance for different rates of condenser coolant flow.

Appendix 6.3 summarises the data obtained from three separate experimental tests on the heat pipe. The data is displayed as Tests A, B and C, arising out of extensive data readings taken during the experiments.

This summary information is included to provide meaningful discussion on the heat pipe characteristics and is specifically shown in this form for a review on heat pipe modulation.

5.2 PRECISION/CONVENTIONAL SYSTEM RESULTS

Similar experimental readings were taken and recorded for the heat pipe, conventional and precision systems.

Appendix 6.4 details the results obtained for the precision and Appendix 6.8 for the conventional system. For these two systems the heat pipe was removed and typical experimental readings are shown below:-

Event time - minutes	Bulk oil temp 8 - °C
Oil heat outlet temp 1 - °C	Coolant outlet temp 9 - °C
Oil heat inlet temp 2 °C	Temp diff water °C
Coolant inlet temp 3 °C	Temp diff oil °C
Valve area temp 4 - °C	Coolant flow cc/min
Valve area temp 5 - °C	Heat flow oil KJ/min
Valve area temp 6 - °C	Heat flow water KJ/min
Valve area temp 7 - °C	

This data again provided the means of establishing heat transfer values and heat flow for the heating oil and coolant water.

The results recorded in the appendices enabled the heat flows to be calculated as shown in chapters 3 and 4, from which the heat transfer values and comparative performance of the concepts were established.

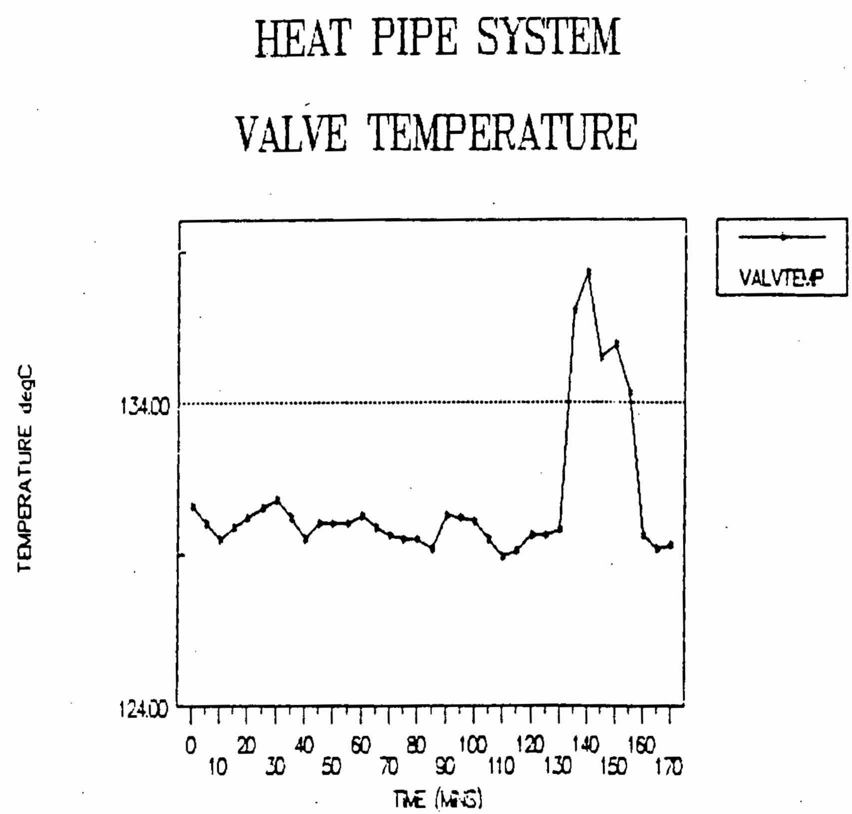
The results obtained were sufficient to establish the potential performance of each concept, maintaining a reasonably constant heating oil inlet temperature and metal temperature in the valve area of the cylinder head. The major variations were induced by changing coolant flow rates and system pressure changes.

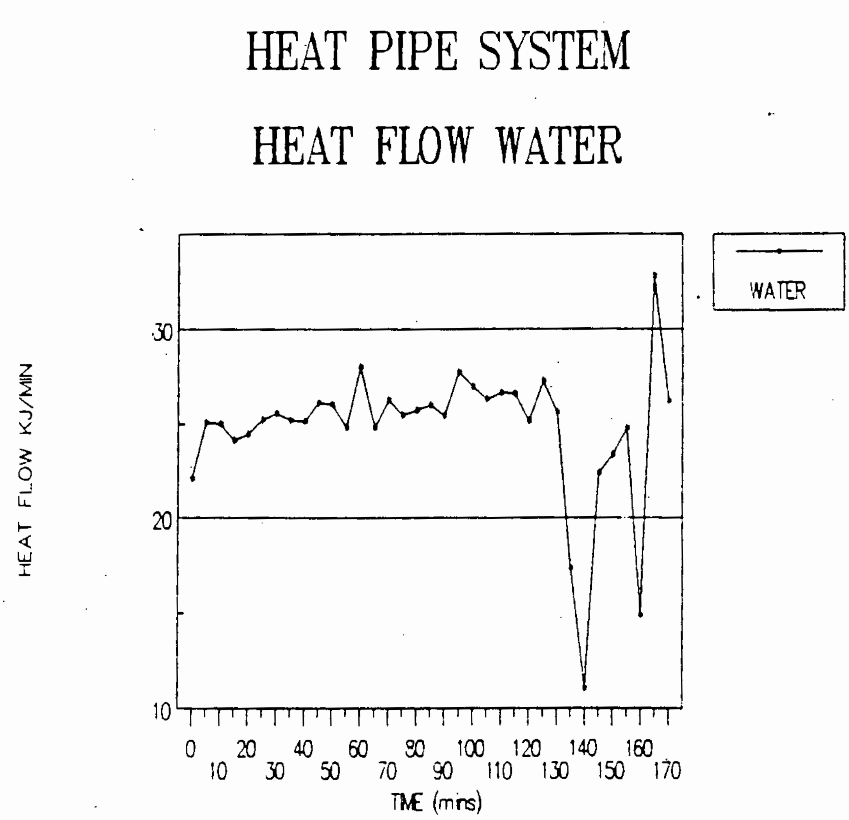
5.3 METAL TEMPERATURES/COOLANT HEAT FLOW TREND

Appendices 6.9 to 6.14 inclusive depict the valve temperatures and the heat flow values for the coolant water over the same period of test for each system.

These appendices clearly show the rapid response of the valve metal temperatures to the variation of coolant flow and heat loss to water. This especially applies to the heat pipe as a system for modulating cylinder head temperature as discussed in chapter 4, section 4.8 and chapter 6.

Typical graphs are shown below:-





CHAPTER SIX

DISCUSSION OF RESULTS

A comparison is made in this chapter of the previous research summarised in chapter 2, the theory proposed in chapter 3 and the results measured in the previous chapter.

This section is divided into the three concepts tested with relevant subsections pertaining to the various tests performed.

INTRODUCTION

The successful use of a system for optimising surface temperature depends on the development of a practical method of improving surface temperature control and on the extent of the effects the improved control produces, thereby improving engine performance and reducing low load exhaust emissions.

Chapters 3, 4 and 5 only concerned heat transfer, under a range of conditions of coolant flowing in various concept configurations. The object being to examine the concepts for improved control of surface temperatures and reduced heat loss to coolant.

6.1 THE HEAT PIPE THERMOSYPHON CONCEPT

6.1.1 Apparatus heating medium - oil

Preliminary tests of the heat pipe performance indicated negative results as seen from chapter 5.

Theoretical calculations for the heating oil medium indicated a flow requirement of approximately 6 litres/min with an assumed oil inlet temperature of 150°C and outlet temperature of 140°C.

Actual oil flow checks were conducted and the apparatus was found to be delivering 6000 cc (6 litres/min) as designed. It was noted, however, that the temperature differential between oil input and outlet was only in the region of 1.9 to 2.0°C and not 10°C as predicted.

The resultant heat transfer to the cylinder head therefore was much lower than the designed input, viz:-

$$Q = \text{Sp heat} \times \text{density} \times \text{temperature diff} \times \text{flow}$$
$$= \frac{2.42 \times 880 \times 1.9 \times 6000}{10^6} = 24.28 \text{ KJ/min}$$
$$= 405 \text{ watts.}$$

As the heat transfer to the apparatus was lower than predicted, the heat pipe operation was sluggish and insensitive. Although the valve bridge and valve temperatures were in the region of 125°C, temperatures in the heat pipe were much lower at 98°C and in the condenser section, the temperatures dropped dramatically. This suggests that low vapour pressure was being developed due to the lower input heat and insuffcient to overcome the pressure at the condenser end, despite varying the condenser coolant flow rate. Fig. 1 and Appendix 6.1 indicate the typical experimental temperatures obtained for two flow rates with the initial heat pipe apparatus tests.

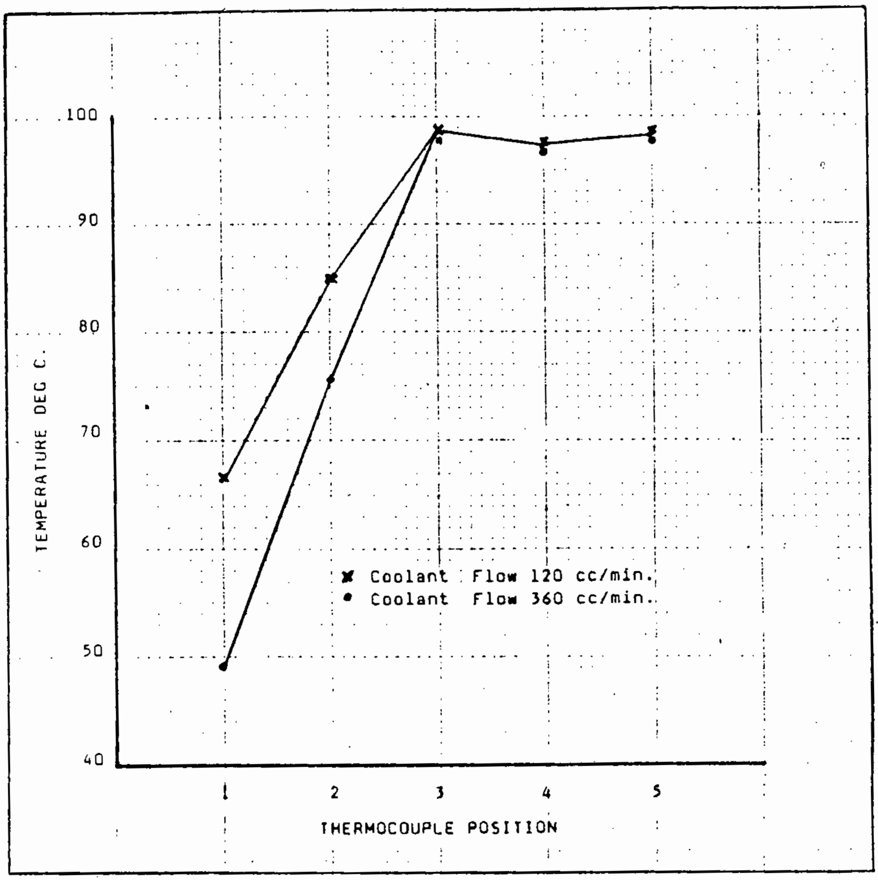


Fig. 1 Temperature Profile Experimental Heat Pipe

The lowered condenser temperature due to the higher coolant flow of 360 cc/min appears to confirm Kemme's theory (Figs. 2 and 3) that choked flow conditions develop as sonic velocity was achieved at the end of the evaporator. Under these conditions the heat pipe operation will not be isothermal as considerable axial temperature and pressure changes will exist under sonic limitation conditions.

This theory is significant as the test results of the heat pipe show similar occurrences under various conditions of test and will be discussed together with the specific results.

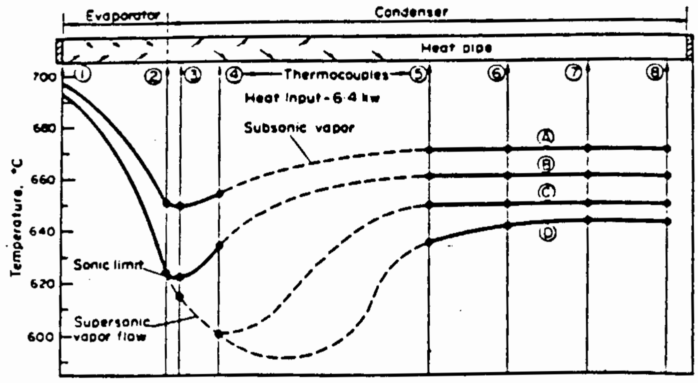


Fig. 2 Temperature profile

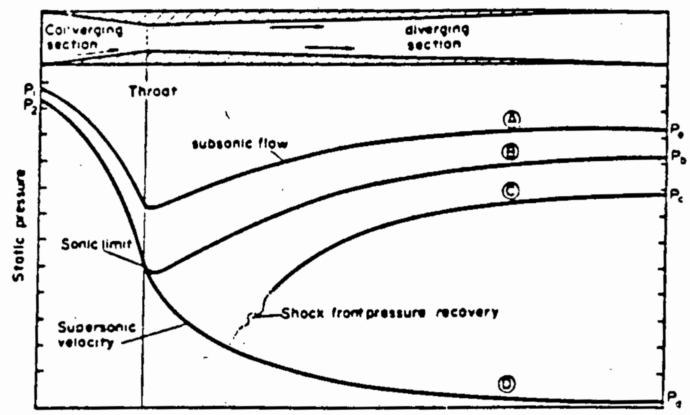


Fig. 3 Pressure profile

In view of the initial indifferent performance of the heat pipe, it was decided to increase the oil flow and the bulk oil temperature. This operation is described in the experimental design section, chapter 4.

The resultant oil flow was raised to 12 litres/min and bulk oil temperature to 175°C.

6.1.2 Heat Input Potential

The new experimental values for oil flow and temperature resulted in a heat input transfer of approximately 900 watts to the heat pipe. The 405 watts achieved with the initial test rig being more than doubled with the modified apparatus..

Reference to chapter 3, section 3.1.7 indicates that the design requirement is 2000 watts or 3810 w-m.

As the length of the heat pipe body is approximately 330 m/m long the required heat transfer is .33 x 3810 = 1257 watts.

The heat pipe length being an approximation due to the integration with the cylinder head, it was assumed that the 900 watts was adequate for the purpose of the experiment. The ideal of 2000 watts not being possible due to non-availability of a suitable oil pump and electric motor to further raise the heat input.

The 900 watt input was is considered satisfactory to demonstrate the heat pipe principle and to develop a comparison of performance with the two other experimental concepts being researched.

6.1.3 Heat Pipe Fluid Inventory

Dunn and Reay[1] stressed the importance of fluid inventory mainly for heat pipes equipped with a capillary wick. If the working fluid inventory is excessive a noticeable temperature gradient can exist at the condenser. Likewise if the fluid inventory volume is too small the pipe fails due to the reduced vapour pressure and filling problems.

In chapter 3, Strel'tsov's equation was used to calculate a proposed fluid inventory volume for the experimental heat pipe. The value calculated being 303 cc.

In the experimental procedure volumes varying from 250 cc to 520 cc were used for the various tests (refer appendix 4.13).

This was done to explore the heat pipe operation and to investigate a bias towards the optimum inventory for continued testing. Also to observe the correlation between the theoretical and the experimental fluid volume.

In general, the higher fluid inventories result in the heat pipe lacking sensitivity and poor response to external variations of system pressure and condenser coolant flow.

The experimental heat response was at its best using the theoretical fluid inventory of 300 cc, producing repeatable results when other factors were varied.

6.1.4 Heat Pipe Performance

The results of various heat pipe tests are detailed in chapter 5. Appendix 6.2 schedules typical test results for the heat pipe tests and displays the temperature readings obtained together with the calculations of oil heat input and water heat output of the apparatus including coolant flow.

The main objective of the experimental work has been covered in chapter 3, on theoretical development. In summary the experimental tests were conducted to observe the overall performance of the heat pipe and its ability to transport heat from the cylinder head in the evaporator area, to the condenser region. The test under controlled conditions, quantifies the extent of the heat extraction from the cylinder head.

The results show the variation in temperatures in the heat pipe for the different values of coolant flowing through the condenser.

Analysis of appendix 6.2 clearly shows that the heat pipe, under the test conditions, has assumed a descending axial temperature gradient trend from outset of the test until the readings taken at time 130 minutes.

Appendix 4.3 indicates the relevant thermocouple positions.

Appendix 6.2 shows that thermocouple Nos. 1 and 2 in the condenser region indicate lower temperatures, due to the sonic choked condition, as previously discussed. The variation in coolant flow viz 208 cc/min - 260 cc/min, not having much apparent effect on the heat pipe temperature gradient.

Reference to Appendix 4.3 shows that thermocouple No. 2 is positioned at the start of the condenser zone. It is probable that the sonic flow choked condition existed in this area during this stage of the experiment.

Time Min	Temp 1	Temp 2	Temp 3	Temp 4	Temp 5	Flow cc/min
130	58.80	86.40	97.60	94.80	94.30	240
135	**44.60**	**41.60**	**67.80**	**82.90**	**94.80**	**232**
140	48.80	91.40	111.20	108.10	107.60	232
145	60.00	93.80	109.00	105.90	105.20	224
150	65.40	93.20	107.40	104.10	103.60	220
155	67.20	92.50	105.90	102.90	102.30	216
160	**95.10**	**90.40**	**97.90**	**95.30**	**94.90**	**108**
165	65.30	83.00	98.10	95.30	94.60	400
170	59.50	83.40	98.10	95.30	94.70	320

Table 6.1 Extract of Appendix 6.2

Thermocouple Position	130 mins Temp °C	135 mins -Temp °C Immediately after Ext. Pres Applied	160 mins-Temp°C Isothermal Condition
1	58.8	44.6	95.1
2	86.4	41.6	90.4
3	97.6	67.8	97.9
4	94.8	82.9	95.3
5	94.5	94.8	94.9

Table 6.2 Extract of Appendix 6.2

From time 130 - 170 minutes, the condition changes as illustrated in **Table 6.1** above. This indicates a dramatic change in the heat pipe temperature gradient as external pressure is introduced to the system by means of compressed air, at time 135 minutes..

The sudden lowering of temperatures extending towards the evaporator end after 130 minutes is explained by the combined effect of introducing cooler air into the pipe and momentary increase of pressure at the condenser end.

As the general pressure rises in the system a rapid recovery from the evaporator end takes place as the potential boiling temperature is increased due to increased pressure and the vapour pressure increases to overcome the sonic choking at the condenser boundary. The heat pipe eventually becoming isothermal at 160 minutes, with balanced but lower temperatures. **Refer Tables 6.1 and 6.2 above.**

Immediately after attaining isothermal conditions at 160 minutes the coolant water flow was increased to 400 cc/min, improving the condenser efficiency but the heat pipe rapidly reverted to sonic choked conditions with a sharp heat differential at the condenser boundary area.
Refer Table 6.1.

Immediately after the heat pipe became isothermal with a flat temperature gradient, consistent up to the coolant end of the condenser, the heat transfer to coolant was at its peak value for the experiment.

Viz: 32.8 KJ/min or 546 watts. **Refer Appendix 6.2.**

The heat input into the apparatus at this time was:

50.5 KJ/min or 842 watts.

$$\therefore \frac{\text{Heat outflow (Water)}}{\text{Heat input (Oil)}} = \frac{546}{842} = 64{,}8\%$$

Within the overall limits of experimental error the oil heat input of 842 watts compares favourably with the 900 watts predicted in section 3.1.2 of this chapter.

The heat removal of 64.8% by the coolant for the heat pipe is higher than the 52% predicted for the conventional system in section 3.2 of chapter 3, possibly due to ideal conditions prevailing at the time of isothermal operation.

In chapters 4 and 5 the objectives for the three heat pipe Tests A, B and C, were outlined and the test results summarised in Appendix 6.3. The main purpose being to provide data on the heat pipe characteristics and the trends developed during the experiments.

The heat pipe performance discussed above, therefore, is based on the results shown in appendix 6.2 and the Test C data extract contained in appendix 6.3.

The phenomena observed for Test C is also confirmed by Test A and B (Appendix 6.3), where it is shown that exactly the same trend exists when the system pressure is varied in the heat pipe.

This trend can be summarised as follows:-

(1) As the system pressure increases in the heat pipe, the condenser temperature decreases initially and exerts an influence over the evaporator region.

(2) The evaporator immediately responds to the higher pressure with temperatures increasing rapidly in this region due to the higher system pressure.

(3) The resulting increased vapour pressure overcomes the sonic choke phenomena discussed in section 3.1.14

(4) The choked condition being overcome allows the heat pipe temperature profile to migrate throughout the heat pipe to a stabilised isothermal condition.

It was also observed that whenever isothermal conditions existed with a relatively flat temperature gradient, that the valve bridge temperature was at its lowest. This indicates that the heat pipe was most efficient in these conditions and supports the heat pipe concept theory.

Condenser coolant flow at isothermal conditions also showed a marked drop, suggesting that some steam vapour was occurring in the coolant and this is confirmed by the increase in temperature differential shown in the experimental results. (Refer Appendix 6.2 - event time 160 minutes.)

The observed results and conclusion reached indicate that the heat pipe has potential to transfer a high proportion of the heat input for the configuration tested. The ultimate required heat transfer capacity of 2000 watts was not achieved due to the apparatus heat input being limited by non-availability of appropriate equipment. This will provide scope for the extended research of the concept at a later stage.

The heat pipe apparatus achieved a heat input of 842 watts and was capable of transferring 64.8% of this input to the water coolant. Its relative performance was therefore adequately demonstrated despite the lower primary heat input values.

The experimental research and results obtained show that a clear potential exists for cylinder temperature modulation by varying the heat pipe system pressure. This confirms the theory outlined in chapter 3 that by means of gas buffering and vapour pressure control, temperatures in the evaporator region can be manipulated. The rapid response identified in the tests introduces a means of modulating the heat performance to achieve a high heat soak condition required at engine idle to reduce emissions and quick response to heat removal through the transient acceleration conditions of acceleration under wide open throttle.

6.1.5 <u>Heat pipe averaged performance</u>

In practice ideal isothermal conditions would be difficult to maintain because of the many permutations of heat input, coolant and pressure. However, the latter two conditions were varied during the experiments and an ideal basis for comparison of the heat pipe with the other proposed concepts would be to consider and compare the averaged results from the data obtained experimentally.

For the heat pipe we can schedule the averaged performance data derived from Appendix 6.2 and displayed in Table 6.3.

Heat input (Oil)	44.97 KJ/min (750 watts)
Heat outflow (Water)	27.58 KJ/min (460 watts)
Percentage heat in/out	61.3%
Valve temperature	130.72°C
Bulk Oil temperature	175.5°C

Table 6.3

These values will be used in a performance comparison when discussing the two other cooling/heat loss concepts.

6.2 CONVENTIONAL COOLING

In chapter 3, the rationale for using a conventional automotive cylinder head as the basis of the experimental apparatus was explained. The main advantage being the ability to apply various cooling concepts to a common base, thereby providing a means for comparing and assessing relative performance under semi realistic conditions.

Thus, for known values of energy input, the experimental results would provide details of energy lost to "exhaust" and energy absorbed by the coolant.

Because of the complex geometry of a cylinder head coolant flow passages, it is difficult to determine the local liquid velocities, as can be done for the small bore precision cooling concept. Therefore average values for coolant and heat flow need to be used for conventional cooling performance assessment.

6.2.1 Predicted Performance

Reference to chapter 3, indicates a predicted performance for the conventional cylinder head apparatus, based on the heat distribution and energy balance figures suggested by Holmes(4).

Viz: For the apparatus heating oil temperature of 175°C the best performance predicted is:-

Coolant Flow	115,2 cc/min
Heat Input	852 watts (oil)
Heat Output	442 watts (coolant)

Ratio Heat in/Heat out = 52%

6.2.2 Experimental Performance Averaged

The actual results achieved are scheduled in appendix 6.8. From the results it is seen that at certain flow rates the predicted values for heat flow are close to those obtained experimentally. Taking the results as a whole, and when averaged the following values are obtained:-

Coolant Flow	103 cc/min
Heat Input	59.90 KJ/min = 998 watts (oil)
Heat Output	31.66 KJ/min = 527 watts (coolant)

Ratio Heat in/Heat out = 52.8%

These results correlate well with the predicted values obtained in chapter 3 (ref: 3.2.2).

6.3 THE PRECISION COOLING CONCEPT

6.3.1 Calculated Heat Transfer - Experimental Velocity

A typical heat transfer prediction for small bore coolant tubes was developed in chapter 3. A theoretical model suggested that for a coolant flow of 200 cc/min and velocity of 5,91 cms/sec, the heat transfer capability of a 6 m/m diameter tube is:-

Heat transfer coefficient h = 766 $w/m^2/°C$

The experimental results and recorded data are shown in Appendix 6.4. Based on these experimental results, coolant velocities and heat transfer coefficients were calculated. Reference to Appendix 6.5 shows good correlation between the experimentally derived calculation and the predicted h = 766 $w/m^2/°C$ above, for similar coolant velocities.

The heat transfer coefficient values for a single tube in Appendix 6.5 varying from 424 $w/m^2/°C$ to 824 $w/m^2/°C$ for velocities ranging from 2.83 cms/sec to 6.49 cms/sec.

The experimentally derived calculations shown in Appendix 6.5 are based on the heat transfer equation proposed by Mansfield and Grover[11] and referred to in chapter 3, section 3.3.1 (Theoretical performance for small bore coolant tubes).

The heat transfer coefficient is given by the equation:-

$$h = \text{constant} \times V^{0.8}$$

The constant value developed in chapter 3 was 7354. Therefore heat transfer coefficient = h = 7354 $V^{0.8}$. Appendix 6.5 was developed on the basis of the constant = 7354 and velocity subject to coolant flow.

This heat transfer range is reflected in Table 6.4 below which summarises the two extremes extracted from Appendix 6.5.

Flow cc/min	Velocity cms/sec	$V^{0.8}$	h Const x V0.8 J/m²/°C/sec	h One Tube Watt/m²/°C	h Two Tubes w/m²/°C
96	2.83	0.0577	424	424	848
220	6.49	0.01121	824	824	1648

Table 6.4 Precision cooling - calculated heat transfer coefficients

From the above it is seen that for the experimental coolant flow velocities, the heat transfer coefficients realised are in agreement with the calculated figures. It now remains to analyse the actual heat transfer achieved by the apparatus during the tests conducted at the various coolant flow velocities.

Appendix 6.5 schedules heat transfer coefficients calculated from the experimental coolant flow readings.

6.3.2 Experimental Heat Transfer

Appendix 6.6 details the apparatus metal and coolant temperatures recorded and the total heat flux in watts and heat transfer coefficients derived from the experimental readings. The basis for the values developed is detailed in Appendix 6.7.

Table 6.5 summarises the salient figures for the velocity and heat transfer coefficient spectrum, these being extracted from the detail in appendix 6.6.

Flow cc/min	Velocity cm/sec	Watts Output	Experimental h $w/m^2/°C$	Calculated h $w/m^2/°C$
70	2.06	364	1680	659
96	2.83	375	1400	848
220	6.49	509	1690	1648

Table 6.5 Precision cooling - Experimental heat transfer coefficients

Reference to table 6.5 and examination of Appendix 6.6 indicates that for the higher velocities of coolant flow the experimental heat transfer coefficient correlates well with the theoretical calculations. The theoretical calculations being based on the experimental velocities.

For the lower flow velocities the small bore tubes showed a marked increase in heat transfer. For some velocity values the experimental heat transfer values were approximately double that of the calculated values.

This variation from the calculated basis for heat transfer is assumed to be the effect of nucleate boiling phenomena.

This assumption is confirmed by the temperature readings recorded during the experimental tests. From appendix 6.4 it will seem that for the lowest flows of 2.07 cm/sec occurring at time 155 minutes onwards the water outlet temperature was at its highest with regions of the metal temperatures showing a sudden marked drop from 144°C to 126.9°C. Refer Appendix 6.4, column temperature 4.

The increased heat transfer of the apparatus at lower coolant velocities must therefore be ascribed to the nucleate boiling theory.

In this experiment the combination of nucleate boiling potential at lower coolant velocities and the heat transfer potential of coolant at higher velocities in the small bore tubes indicates that an almost consistent heat transfer capability of approximately 1500 - 1700 $w/m^2/°C$ is possible with the given heat input.

6.3.3 Nucleate Boiling - Effect on Precision Cooling

The nucleate boiling theory was outlined in chapter 2 and in chapter 3 Theoretical Development: an opportunity was identified to explore this phenomena with the precision cooling system. By controlling the coolant flow and velocity, the passage pressure is reduced and the coolant boiling point lowered. Boiling potential is increased and in the phase change from liquid to vapour, the coolant absorbs a considerable amount of heat, thereby enhancing the heat transfer.

Whilst chapter 2 indicated that extremely high transfer rates could be achieved with stable and ideal nucleate boiling conditions, the experimental results recorded in appendix 6.4 and 6.6 show that the heat transfer values were only increased by 2 to 3 times the calculated values when this occurred. For example, at event time 165 minutes the experimental transfer value peaked, achieving 1820 $w/m^2/°C$ and the calculated value 650 $w/m^2/°C$. This suggests that the nucleate boiling condition only existed momentarily and that the consistent high heat transfer potential of stable nucleate boiling was not achieved.

At this stage of the experimental tests the coolant flow was observed to be erratic with small pockets of steam emerging.

In previous chapters it was shown heat flux in nucleate boiling cannot be increased indefinitely.

Reference to Fig. 4 indicates that a vapour film forms as a result of the vapour bubble population. The condition of "blanketing" then occurs.

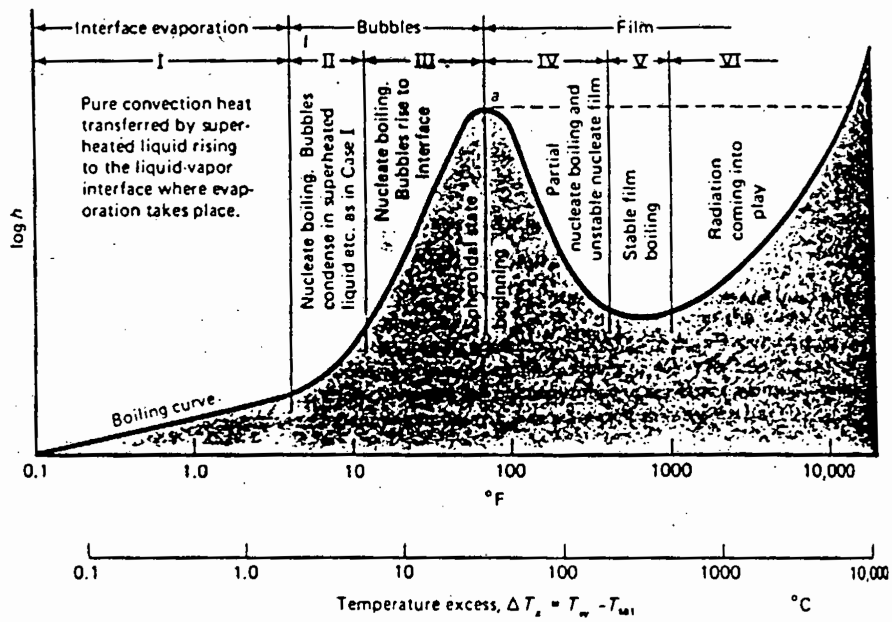


Fig. 4

In the experimental tests, the anticipated lower heat transfer values predicted by the calculation therefore, did not materialise at lower coolant velocities for the above reasons.

It appears that a degree of nucleate boiling at the <u>lower coolant velocities</u> assisted in maintaining a high heat transfer rate with stable metal temperatures.

Mansfield and Grover[11], page 12, indicated that under nucleate boiling conditions, the resultant raising of passage surface temperatures did not increase the heat transfer coefficient, which remained constant.

The average results are scheduled in Table 6.6 below:-

Concept	Heat In Oil KJ/min	Heat Out Water KJ/min	Heat Out Water Watts	Ratio Water/Oil %	Av.Valve Temp °C	Av.Bulk Oil °C
Conventional	59,97	31,66	527	52,8	127,0	177,4
Heat Pipe	44,97	27,58	460	61,3	130,7	175,5
Precision	35,12	24,86	414	70,7	132,3	177,2

Table 6.6 Concept averaged results

Before comparing the various concept results it is necessary to explain the basis for the "heat in" column in Table 6.6.

This column is derived from the heating oil temperatures recorded during the relevant tests. The oil flow rate and bulk oil temperatures being stable and consistent throughout the experiments. With consistent oil inlet temperatures the heat flow values for the "heat in" column will vary according to the outlet temperatures. Therefore a higher "heat in" (heat retained) value reflects a lower oil outlet temperature, which in turn suggests a larger differential between the oil IN and oil OUT temperature and a higher heat transfer into the apparatus.

6.4.1 Conventional System Comparison

For example, the "heat in" value of 59.97 KJ/min for the conventional concept, being higher than 44,97 KJ/min for the heat pipe heat inflow, indicates that a smaller heat differential was established by the heat pipe and that hotter oil was exhausted and therefore more heat energy returned to the bulk oil tank.

Consider the analogy, that the heating oil medium, used in the experiments, be regarded as the combustion and exhaust gas flowing through the cylinder head. The average results contained in Table 6.6, then provide an interesting interpretation and comparison of the potential performance and capability of each concept examined.

The conventional concept is used as a data base for analysing Table 6.6.

Reference to Table 6.6 indicates that highest heat retention in the cylinder head was achieved by the conventional concept. Viz 59,97 KJ/min, with a corresponding highest heat rejection to coolant of 31,66 KJ/min, equating to the coolant only extracting 52,8% of the heat input.

These figures therefore indicate that the conventional has the lowest efficiency of the three concepts. This confirms the earlier discussion, section 6.2, when it was shown that the experimental results obtained for this concept agree with the research and theoretical predictions for heat loss and energy distribution.

According to the assumed analogy this means that for the conventional concept, the least heat is transferred to the exhaust and the most transferred to the coolant water, therefore having the highest heat loss of the three concepts.

This then suggests that the remaining two concepts are potentially more efficient.

6.4.2 The Heat Pipe Comparison

The average results recorded for the heat pipe indicate that 61,3% of the heat input was removed by the condenser coolant flow, as opposed to 52,8% and 70,7% of the conventional and precision concepts respectively.

Accepting the principle that the experimental heating oil simulates the combustion and exhaust gas energy then it is seen that 25% more energy is released to "exhaust", that is 44,97 KJ/min as against 59.97 KJ/min for the conventional concept.

6.4.3 Precision Cooling Comparison

Table 6.6 shows that 70,7% of the heat input was removed by the coolant for this concept. Thus confirming that this system is the most efficient in terms of heat transfer. The heat pipe recording 61,3% and the conventional system only achieving 52,8% heat outflow. By examining the heat-in values, these show that 20% more energy is passed to the exhaust on a simulated basis when compared with the heat pipe and 40% more than the conventional system.

With all three of the concepts the average results show reasonably consistent valve metal and bulk heating oil temperatures.

The overall analysis of the three concepts indicates that the precision cooling concept is the most efficient, followed by the heat pipe, with the conventional system the worst performer.

A further general observation of the experiments and the results shows that the metal temperatures of the apparatus were reasonably sensitive. A good response to coolant flow variation was noticed for all the tests. Appendices 6.9 and 6.10 show a graphic representation of the heat flow and valve temperature for the precision concept tests, these also being typical for the other concepts.

All readings being taken simultaneously show that the metal response is in sympathy with the coolant heat transfer values.

CHAPTER SEVEN

CONCLUSION

The heat loss tests provided an insight into the heat transfer phenomena as applied to the three systems investigated. This work also illustrated the effects of the variation of coolant flow and velocity and the influence of nucleate boiling. In particular the variables applied to the heat pipe system provided interesting data which could be explored for future development of this system.

7.1 THE HEAT PIPE/THERMOSYPHON

7.1.1. Initial Design

The initial experimental heat pipe design was selected on the basis of physical size limitation and practicality for small internal combustion engines. The calculated results were found to be totally inadequate to transfer the required heat flow. The selected dimensions of the copper heat pipe were 12 mm diameter by 20 cm long. In depth calculations (section 3.1.7) indicated a maximum heat transfer factor of 138,5 w/m whereas the requirement to dissipate the required 2000 watts was 3810 w/m. Authorities such as Dunn & Reay[1] confirmed equivalent low values for similar sized pipes as the initial design.

7.1.2 Integrated Design

An integrated cylinder head/heat pipe configuration was chosen to simulate the heat pipe system for further experimental tests using the cylinder head water jacket as the evaporator area. The plant-on heat pipe calculated in 3.1.7 being abandoned.

Despite the reservations on impurities in cylinder head castings identified in sub-section 3.1.11, chapter 3, the integrated system functioned according to the anticipated heat pipe/thermosyphon characteristics, displaying all the theoretical aspects detailed in the previous chapters.

7.1.3 Apparatus Heat Input

The temperature gradients and differences assumed for the test apparatus did not materialise in the initial tests due to the inadequate heat inflow of the oil heating medium. In chapter 3 "Theoretical Development" a heat input of 2000 watts to the apparatus was predicted with an oil flow of 6 litres/minute. Only 405 watts was achieved, mainly due to low oil flow and bulk oil temperature. Modifications were made to the apparatus and subsequently the heat input was increased to approximately 900 watts, being the limit of the apparatus equipment available. This was considered satisfactory to demonstrate the relative performance of the systems to be tested.

7.1.4 Sonic Choked Conditions

The heat pipe is sensitive to sonic choked conditions at the evaporator condenser interface. This phenomena was explained in chapters 3 and 6 and negatively affects the performance of the heat pipe. Factors which influence this condition are:-

(a) Condenser coolant temperatures.
(b) Evaporator heat-in flow temperatures.
(c) System pressure.

Vapour compressibility and flow therefore play a vital role on the performance of the heat pipe. As suggested by Kemne in chapter 3, section 3.1.14, lowering the condenser temperature induces sonic choked flow conditions.

Alternatively reducing system pressure or evaporator heat input extends the condenser temperature towards the evaporator region.

Under sonic choked flow conditions the heat pipe operation will not be isothermal as considerable axial temperature and pressure changes will exist.

This phenomena was clearly shown under experimental conditions and discussed in chapter 6 with reference to Tables 6.1 and 6.2.

By increasing the evaporator heat input or increasing system pressure to raise evaporator temperature, the flow choked condition can be overcome. This allows the heat pipe to become isothermal with a flat temperature gradient, consistent up to the coolant end of the condenser.

Under isothermal conditions the heat pipe produced its maximum heat flow. In this case 546 watts when in the isothermal state against 460 watts under average operating conditions. A 19% increase over average.

7.1.5 Heat Transfer Modulation

Manipulation of the heat pipe performance is possible with superimposed pressure variation as shown above. Therefore it has scope for being modulated to vary heat transfer.

The results and observations made during experimentation show, that the rapid response times during modulation indicates its potential to control cooling and heat loss depending on engine power demand and emission requirements.

This modulation capability addresses one of the objectives identified in the project introduction.

7.1.6 Performance and Limitations

The heat pipe performance shows good potential in relation to the two other systems assessed. It was the second best system in terms of heat loss to heat input 61,3%.

In relation to automotive applications its size limitation may hinder its development. Nevertheless, it would appear that it has potential if developed as an integral unit along similar lines to the experiment.

7.2 THE CONVENTIONAL SYSTEM

7.2.1 Heat Distribution

In chapter 2, heat distribution and energy balance figures for an automotive engine were suggested by Holmes[4]. Based on these values, a predicted heat transfer performance for the conventional cylinder head apparatus was developed in chapter 3.

7.2.2 Predicted vs Actual Performance

The actual average results achieved experimentally correlated well with the theoretical values established in chapter 3, as seen below:-

	Predicted	Actual
Coolant flow cc/min	115	103
Heat input oil watts	852	998
Heat output water watts	442	527
Ratio heat in/heat out %	52,0	52,8

7.2.3 Basis for Comparison

This system performed the worst of the three, heat loss to heat input being the lowest at 52,8%.

The conventional system forms a basis for examining the relative performance of the other two systems.

7.3. PRECISION COOLING

7.3.1 Predicted Performance

The heat transfer prediction for typical small bore coolant tubes was developed in chapter 3. This suggested that for an assumed coolant flow of 200 cc/min and velocity 5,9 cm/sec with a 6 mm diameter tube the:-

heat transfer coefficient $h = 766\ w/m^2/^\circ C$.

7.3.2 Calculated Performance

Calculated heat transfer coefficients, based on the experimental coolant velocities, are scheduled in appendix 6.5 and show good correlation with the above. For coolant velocities ranging from 2,06 cm/sec to 6,49 cm/sec the heat transfer coefficient varied from 424 to 824 $w/m^2/°C$. The predicted coefficient being 766 w/m2/°C at 5.91 cm/sec.

7.3.3 Actual Performance

The actual heat transfer coefficients achieved in the experiments, however, indicate that whilst the values at higher velocities agree with the theoretical calculations, the lower velocity experimental heat transfer values were higher than calculated. Reference to appendix 6.6 confirms this.

Table 6.5 taken from chapter 6 and repeated below, summarises the coefficient spectrum extracted from appendices 6.5 and 6.6.

Flow cc/min	Velocity cm/sec	Watts Output	Experimental h $w/m^2/°C$	Calculated h $w/m^2/°C$
70	2.06	364	1680	659
96	2.83	375	1400	848
220	6.49	509	1690	1648

Table 6.5

7.3.4 Nucleate Boiling Effect

For low coolant velocity values the heat transfer was approximately double that of the calculated values. Referring to appendix 6.4 it is seen that at some stages of the experiment and low water velocities, water outlet temperatures rose sharply and the metal temperature dropped suddenly.

It is concluded that the increased heat transfer of the apparatus at lower coolant velocities must be due to nucleate boiling phenomena, enhancing heat transfer capability.

7.3.5 Heat Transfer Capability

Table 6.5 shows that the heat transfer coefficient for this experiment should have varied from 659 $w/m^2/°C$ to 1648 $w/m^2/°C$ whereas it achieved a reasonably consistent level of 1400-1700 $w/m^2/°C$.

The conclusion is that the experimental results for precision cooling show a consistent heat transfer capability. This is possible, if the heat transfer potential, at higher velocities in the small bore tubes, could be combined with a stable nucleate boiling condition at the lower coolant velocities.

7.4 SYSTEM COMPARISON

7.4.1 Relative Heat Transfer Values

The three concepts have been tested to examine heat losses of a typical automotive cylinder head.

The tests were conducted to assess the heat transfer potential for each system, thereby arriving at relative values for heat loss to coolant and determination of energy lost to exhaust. The latter being possible because of the simulated "combustion and exhaust gas energy" developed by the apparatus oil heating medium and the former calculated from the temperature differential recorded for the coolant flow.

For ease of reference, the average performance results are shown in Table 6.6, taken from chapter 6 and repeated below:-

Concept	Heat In Oil KJ/min	Heat Out Water KJ/min	Heat Out Water Watts	Ratio Water/Oil %	Av.Valve Temp °C	Av.Bulk Oil °C
Heat Pipe	44,97	27,58	460	61,3	130,7	175,5
Conven-tional	59,97	31,66	527	52,8	127,0	177,4
Precision	35,12	24,86	414	70,7	132,3	177,2

Table 6.6

Table 6.6 confirms the conclusions reached by Mansfield and Grover(11) that the precision cooling system located in the critical areas, displays a reduced flow of heat to coolant (414 watts) yet maintaining constant metal temperature.

7.4.2 Precision Cooling Rating

The overall conclusion after reviewing and comparing the relative performance of each system, is that the precision cooling system has the most efficient heat transfer potential.

The heat input retained was the lowest 35,12 KJ/min, that is more heat energy was passed to "exhaust" (heated oil/exhaust gas analogy), whilst the highest <u>percentage</u> transfer to coolant, 70,7%, was achieved. Therefore demonstrating its heat transfer efficiency.

7.4.3 Heat Pipe Rating

The heat pipe system tested was the next most efficient with 44,97 KJ/min heat input retained, i.e. 25% more than the precision system, whilst 61,3% heat input was transferred to the coolant water.

7.4.4 Conventional System Rating

The conventional system retained the highest percentage of heat input at 59,97 KJ/min and only transferring 52,8% to coolant.

7.4.5 Relative Rating

In terms of heat loss the final conclusion is that the precision cooling system is the most efficient in conserving energy, with low heat retention and higher rejection to "exhaust" where it could be recovered, the other two systems being 25% and 32% worse.

7.4.6 Heat Absorption

For heat absorption the precision system is also concluded as the most efficient, being able to dissipate 70,7% of the heat into coolant against 61,3% and 52,8% of the heat pipe and conventional systems respectively.

7.4.7 Application

The **precision cooling system** would permit cylinder heads which would be simple to cast, eliminating the complex geometry of conventional heads and requiring fewer internal cores. This would encourage more maintenance. Easier inspection and cleaning of coolant system also improve casting quality and reduce costs.

The improved efficiency and reduced cost aspects of this system addresses one of the key issues identified at the initiation of this project.

The **heat pipe** clearly shows merit for automotive and internal combustion engine use. Its application, however, is limited by physical size. The performance of the smaller heat pipe could be improved, but will depend on the wick capability and liquids used.

Much more experimentation and research will be required in these areas to produce heat pipes capable of transferring the heat flux required in confined area applications.

The heat pipe experimental results demonstrate its potential to control heat loss and therefore can be considered as an alternative system for cooling small internal combustion engines subject to the further development outlined.

CHAPTER EIGHT

RECOMMENDATION

The discussion and conclusion in the previous chapters show the potential of the systems tested, for controlling heat losses in internal combustion engines.

The research work done on this project has created a data base for further in depth evaluation and development of the heat pipe and precision cooling system. The experimental work also identifies several potential applications for energy and heat loss/cooling control.

In the automotive field particularly, there are vehicle systems which could be considered for heat pipe application, viz:

(i) SUPPLEMENT CONVENTIONAL COOLING SYSTEM
(ii) TURBOCHARGER COOLING
(iii) ENGINE/TRANSMISSION OIL COOLING
(iv) MODULATION OF ENGINE TEMPERATURES
(v) ENGINE INTERNAL COOLING

The precision cooling system could be applied to:

(i) SIMPLIFY CYLINDER HEAD CASTINGS
(ii) SUPPLEMENT CONVENTIONAL COOLING SYSTEM
(iii) COMBINE WITH HEAT PIPE FOR IMPROVED HEAT TRANSFER CONTROL

It is clear that the reduction in heat flow to coolant that can be obtained with acceptable temperature conditions will vary greatly according to engine design and operating conditions.

Based on the findings of the experimental work done on this project, the development and testing of any revised proposals and applications such as the above, will need to be finally tested on complete engines.

If, however, a further test rig programme is envisaged, then several modifications and refinements are necessary for the existing experimental apparatus.

(i) Improved and high accuracy flow and temperature measuring equipment should be obtained. Continuous flow measurement of the heated oil and the coolant will provide more consistency and reduce experimental error.

Similarly temperature recordings should be registered on a multi-channel recorder. This will identify the experimental trends at a glance, facilitate apparatus adjustments and improve consistency of results.

(ii) Higher heating medium temperatures should be attempted to simulate temperatures closer to combustion values. The existing Shell Thermia D oil used is capable of operating at 250°C, but to be used with caution . Fume was evident at 175/180°C causing some physical distress with prolonged testing. This could be a health hazard for the operator unless laboratory ventilation and heating is improved. An alternative heating medium, steam or gas, can also be considered.

(iii) Coolant flow consistency could be improved by using a closed circuit water pump and control valve system. This would be an advantage, subject to strict system temperature control.

A final recommendation is that from the results of the experimental work and the conclusions reached, consideration should be given to the pursuing three projects which show good potential for possible long term application.

(i) A multiple heat pipe arrangement integrated with the cylinder head, as opposed to the single pipe investigated in this project. Physical size limitations and balanced proportions to be observed for practical application.

(ii) An in depth investigation of a pressure modulation system for the heat pipe. This could either be automatically controlled by engine operating conditions or manually by the vehicle operator.

(iii) Design and fabrication of a precision cooled cylinder head to optimise the efficiency of a small internal combustion engine for rural power applications, as reviewed in the introduction to this project

The above concepts, if suitably refined, should have commercial application potential.

CHAPTER NINE

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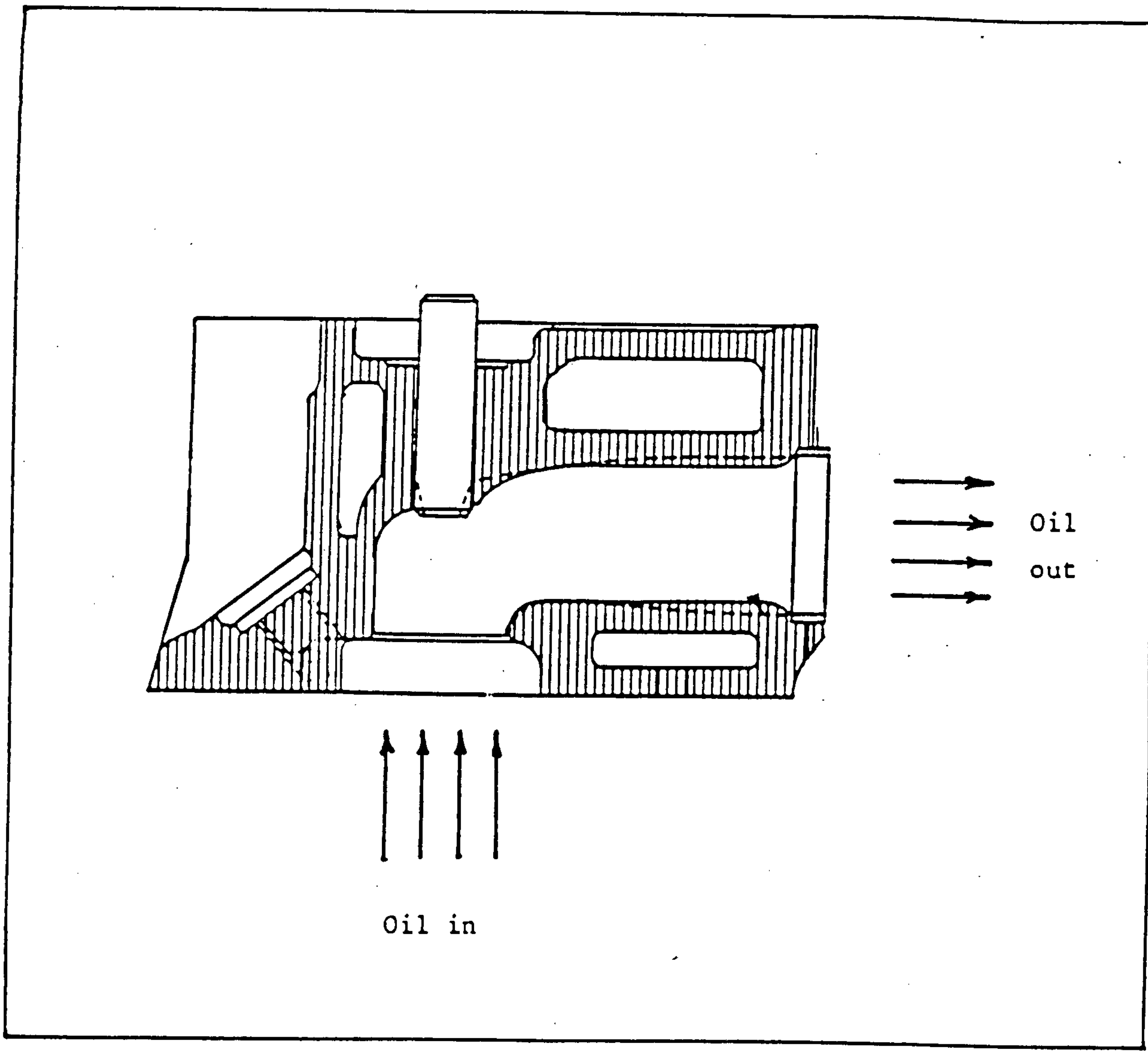
APPENDIX SCHEDULE

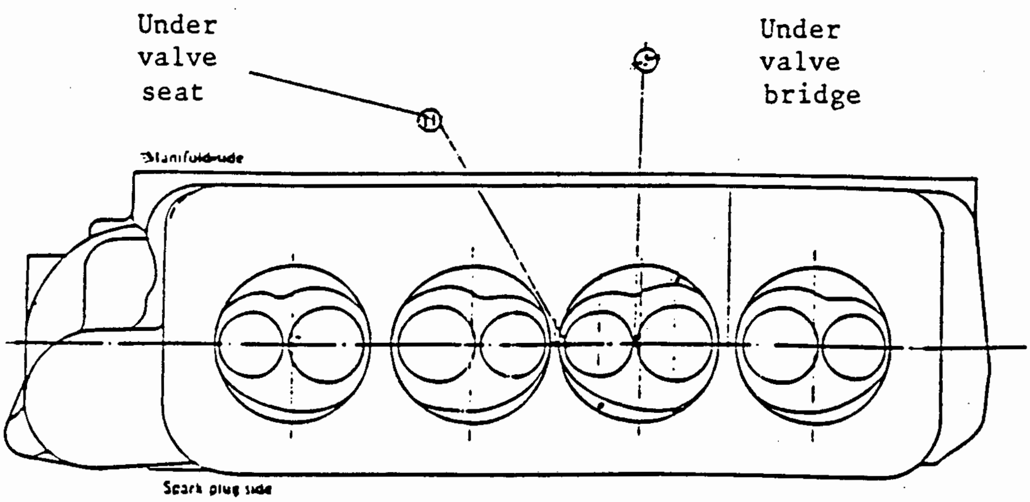
CHAPTER 6

APPENDIX 3.1



CYLINDER HEAD TEST RIG
HEATING OIL FLOW ROUTE

APPENDIX 3.2



LOCATION OF THERMOCOUPLES IN CYLINDER HEAD

APPENDIX 3.3

WICK DESIGN

(REFER CHI[2], PAGE 192)
(NOTE: CHI GRAPHS IN IMPERIAL UNITS)

Assume that the heat pipe condenser elevation is 25 mm above the evaporator, its total length 20 cm and the I D 11,28 mm.

To transfer heat at 125°C, it is required to overcome the hydrostatic height (h).

$$\text{Therefore } h = 25 + 11{,}28 \, \frac{(200^2 - 25^2)}{200^2}$$

$$= 36{,}10 \text{ mm.}$$

Since the hydrostatic pressure per vertical 300 mm at 125°C is 3,00 kpa, the pipe is required to overcome the hydrostatic pressure. (Appendix 3,9, Fig. 7-6.)

$$\text{Hydro pressure} = \frac{36{,}10}{300} \times 3{,}00$$

$$= 0{,}36 \text{ kpa}$$

The wire screen can now be chosen, so that the maximum capillary pressure = 2 x 0,36 kpa = 0,72 kpa.
Reference to Appendix 3.9, Fig. 7-5c indicates:-

Wire screen mesh = 100.

The required wick thickness can now be determined by the equation:

$$\text{Capillary Press Pcm} = \frac{\text{Surface tension coefficient}}{\text{Capillary radius}} = \frac{2\,\sigma}{r_c}$$

and heat transport factor (QL)c

$$(QL)_c max = \frac{Pcm - \Delta P_{\perp} - \rho_1 gLtSIN\Psi}{F_1 + F_v} = \frac{0,456}{F_1 + F_v}$$

At vapour core diameter $dv = 9,52$ mm
and Temperature $T_v = 125°C$
The frictional coefficient
vapour flow $F_v = 1,693$ N/m²/w/m

Refer Appendix 3.10, Fig. 7-8c.

WICK CROSS SECTION

Wick cross section $Aw = \frac{\pi}{4} \times 36,61 = 28,765 \times 10^{-6} m^2$

With mesh N° $= 100$ and $T_v = 125°C$
(Appendix 3.10, Fig.7-7c)

Value of $F_1 A_w = 4,61 \times 10^{-5}$ N/Wm

Now $F_1 = \frac{F_1\,Aw}{Aw} = \frac{4,61 \times 10^{-5}}{28,765 \times 10^{-6}} = 1.6$N/m²/Wm

Now substituting the values:

$$F_v = 1{,}693 \text{ N/m}^2\text{/Wm}$$

$$F_l = 1{,}60 \text{ N/m}^2\text{/Wm}$$

$$Pcm - \Delta P_{\perp} - \rho_1 gLtSIN\Psi = 0{,}456 \text{ kpa}$$

Into

$$(QL)_c max = \frac{Pcm - \Delta P_{\perp} - \rho_1 gLtSIN\Psi}{F_l + F_v}$$

$$= \frac{0{,}456 \times 10^3}{1{,}60 + 1{,}693}$$

$$= 138{,}5 \text{ Wm}$$

For the design under consideration:

$$\text{Required QL} = \text{vapour core dia } (L_e + L_c) \times Q$$

$$= 9{,}52 \ (0{,}1\text{m} + 0{,}1\text{m}) \times 2000$$

$$= \underline{3810 \text{ Wm}}$$

Designed QL = $\underline{138{,}5 \text{ Wm}}$ - therefore inadequate.

APPENDIX 3.4

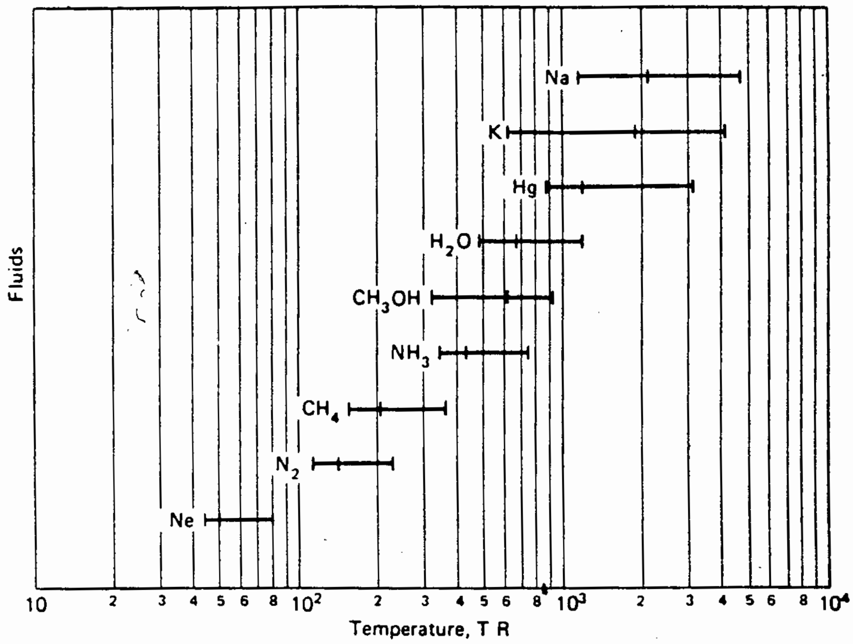


FIG. 6-1 Normal melting point, boiling point, and critical point temperatures for several heat pipe working fluids (1R = 0.5556K).

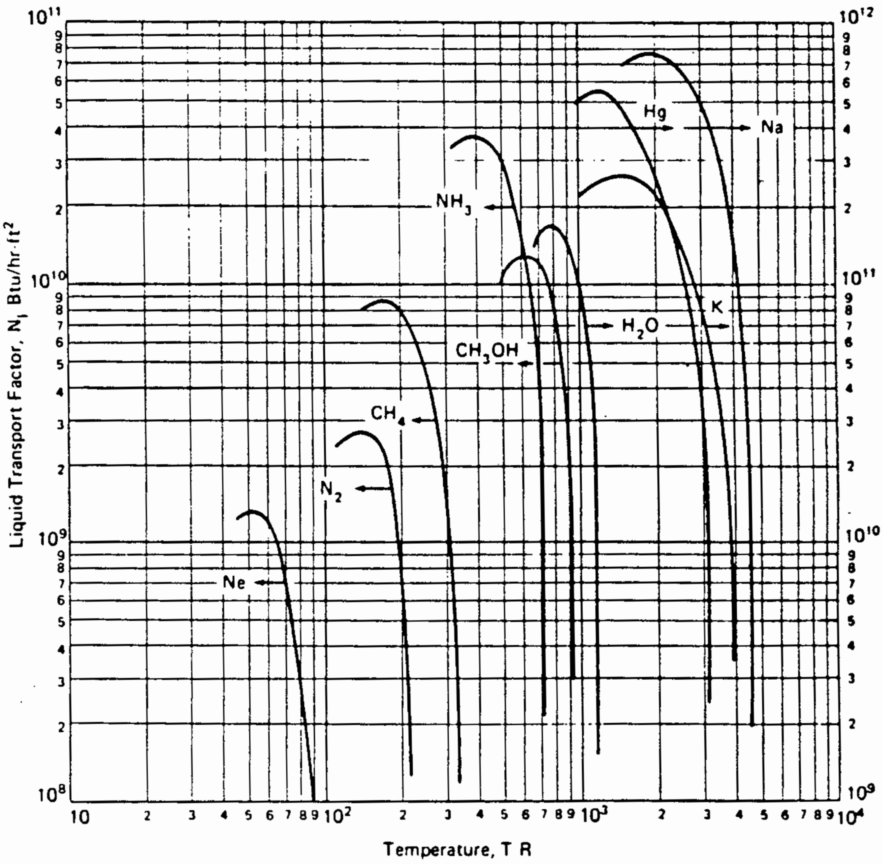


FIG. 6-2 Liquid transport factor versus temperature for several heat pipe working fluids (1 Btu/hr-ft^2 = 3.153 W/m^2; 1R = 0.5556K).

APPENDIX 3.5

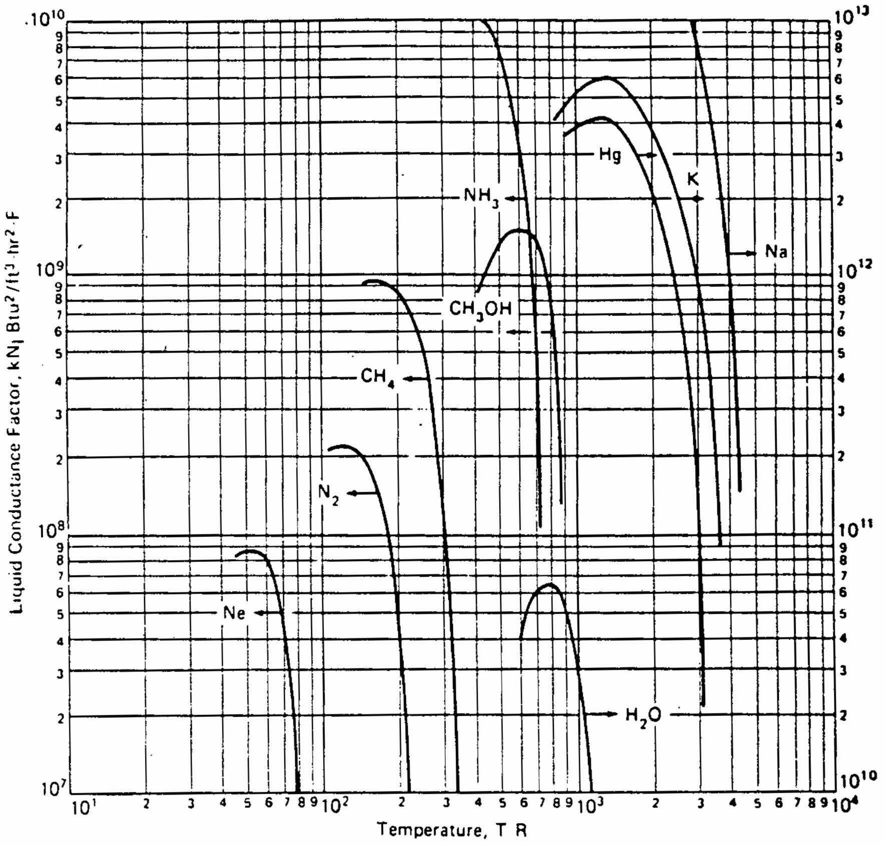


FIG. 6-3 Liquid conductance factor versus temperature for several heat pipe working fluids (1 Btu²/ft³-hr²-F = 5.455 W²/m³-K; 1R = 0.5556K).

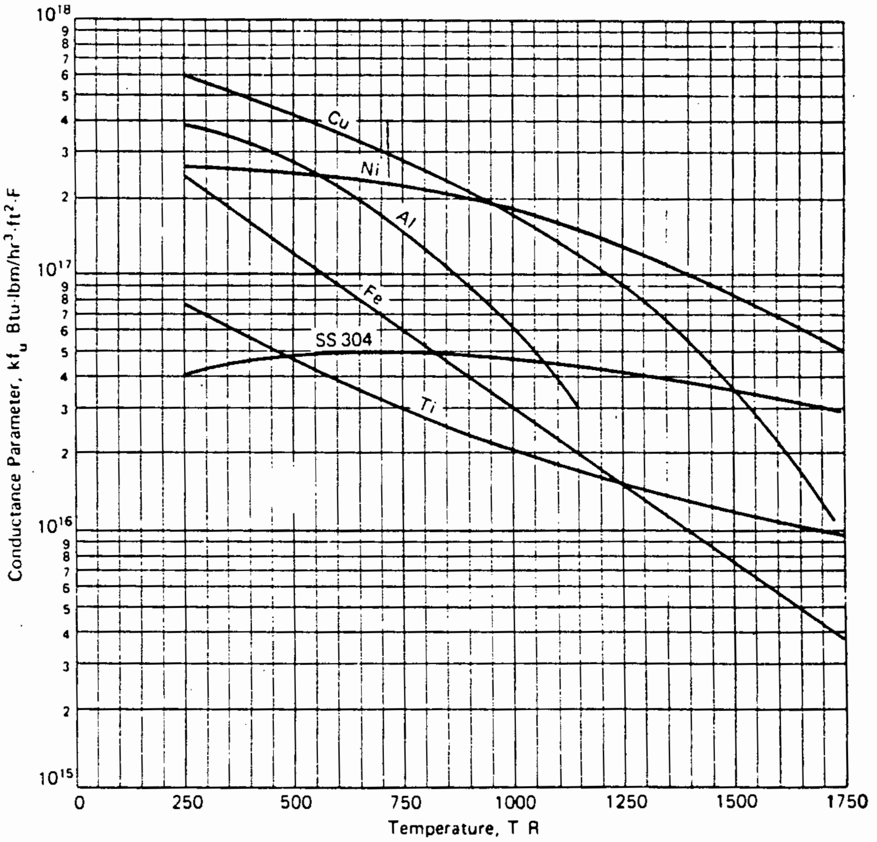
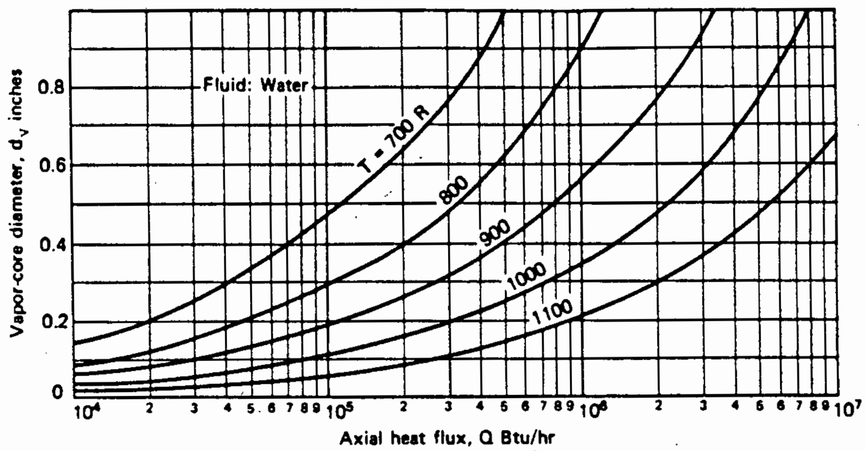


FIG. 6-7 Material conductance parameter versus temperature for several heat pipe materials (1 Btu-lbm/hr³-ft²-F = 1.986 × 10⁻⁷ W-kg/sec²-m²-K; 1R = 0.5556K).

APPENDIX 3.6



(c)

Fig. 7-1

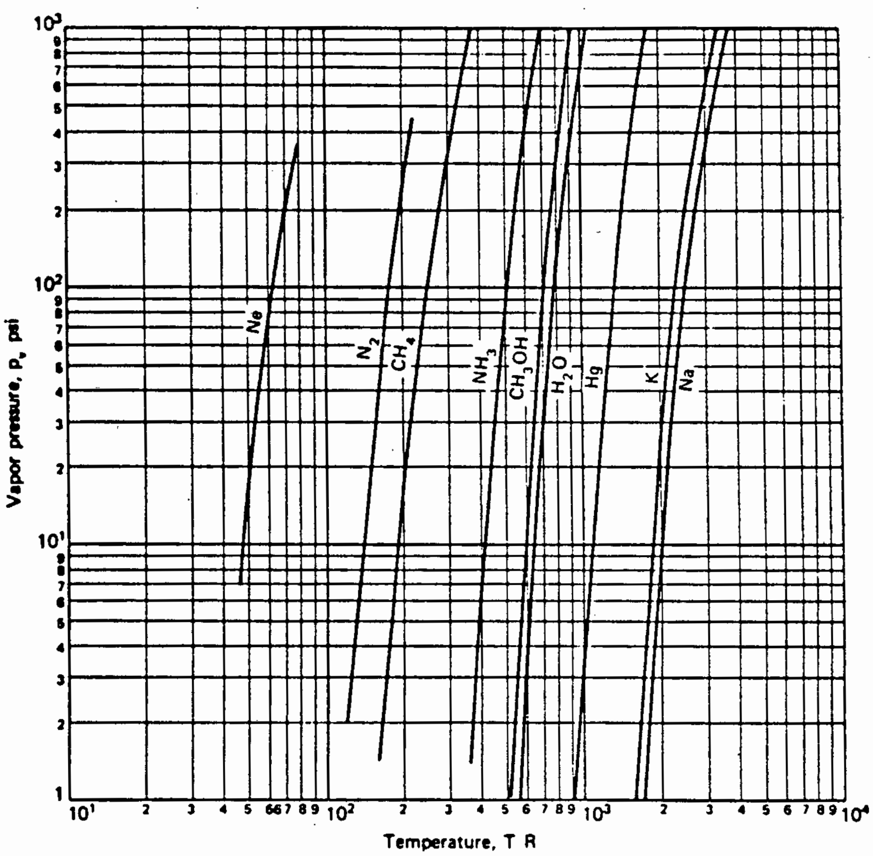


FIG. 7-2 Vapor pressure versus temperature for several heat pipe working fluids ($1\ psi = 6.895 \times 10^3\ N/m^2$, $1 R = 0.5556 K$).

APPENDIX 3.7

TABLE 7-1 Data for Sample Commercial Tubes (1 in. = 0.0254 m)

Tube o.d., in.	o.d./i.d.	Bwg[a]	Thickness, in.	i.d., in.
$\frac{1}{4}$	1.289	22	0.028	0.194
	1.214	24	0.022	0.206
	1.168	26	0.018	0.214
$\frac{3}{8}$	1.354	18	0.049	0.277
	1.233	20	0.035	0.305
	1.176	22	0.028	0.319
	1.133	24	0.022	0.331
$\frac{1}{2}$	1.351	16	0.065	0.370
	1.244	18	0.049	0.402
	1.163	20	0.035	0.430
	1.126	22	0.028	0.444
$\frac{5}{8}$	1.536	12	0.109	0.407
	1.362	14	0.083	0.459
	1.263	16	0.065	0.495
	1.186	18	0.049	0.527
	1.126	20	0.035	0.555
$\frac{3}{4}$	1.556	10	0.134	0.482
	1.410	12	0.109	0.532
	1.284	14	0.083	0.584
	1.210	16	0.065	0.620
	1.150	18	0.049	0.652
	1.103	20	0.035	0.680
$\frac{7}{8}$	1.441	10	0.134	0.607
	1.332	12	0.109	0.657
	1.234	14	0.083	0.709
	1.174	16	0.065	0.745
	1.126	18	0.049	0.777
	1.087	20	0.035	0.805
1	1.493	8	0.165	0.670
	1.366	10	0.134	0.732
	1.279	12	0.109	0.782
	1.199	14	0.083	0.834
	1.149	16	0.065	0.870
	1.109	18	0.049	0.902
	1.075	20	0.035	0.930

[a] Birmingham wire gauge.

APPENDIX 3.8

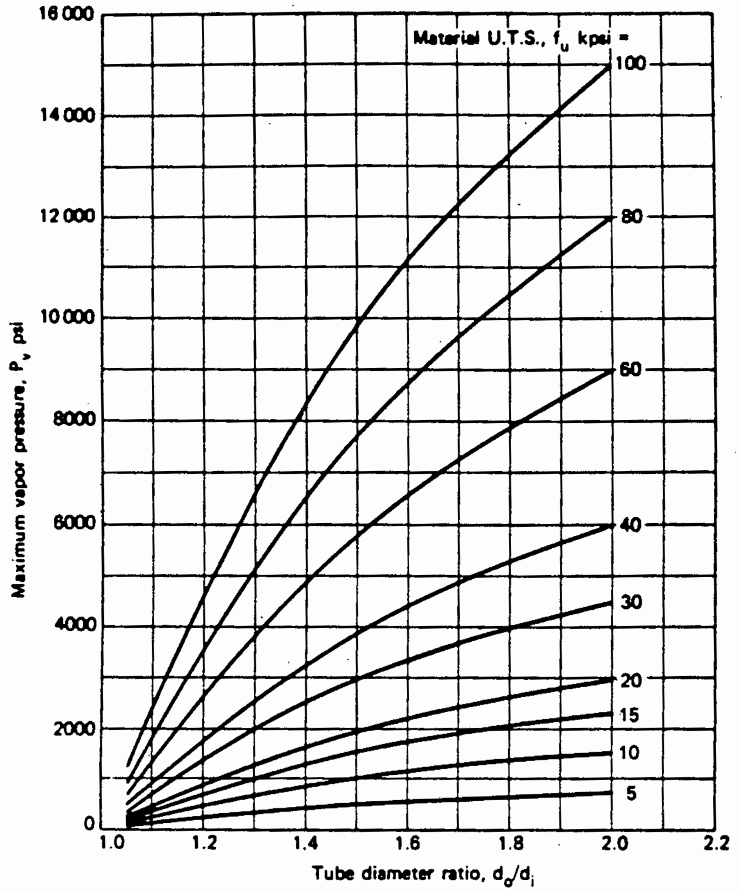


FIG. 7-3 Design chart for heat pipe container tubes (1 psi = 6.895×10^3 N/m², 1 kpsi = 6.895×10^6 N/m²).

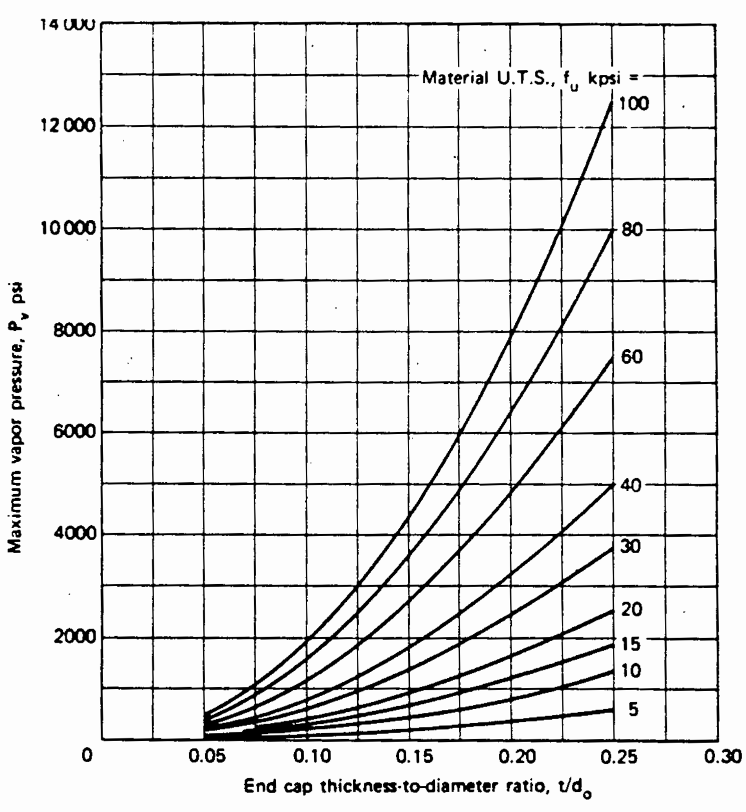
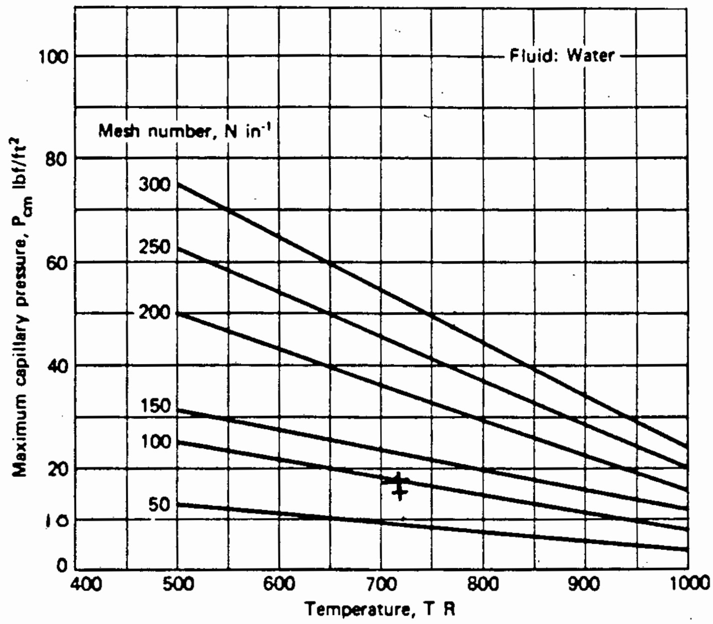


FIG. 7-4 Design chart for heat pipe end caps (1 psi = 6.895×10^3 N/m², 1 kpsi = 6.895×10^6 N/m²).

APPENDIX 3.9



(c)

Fig. 7-5

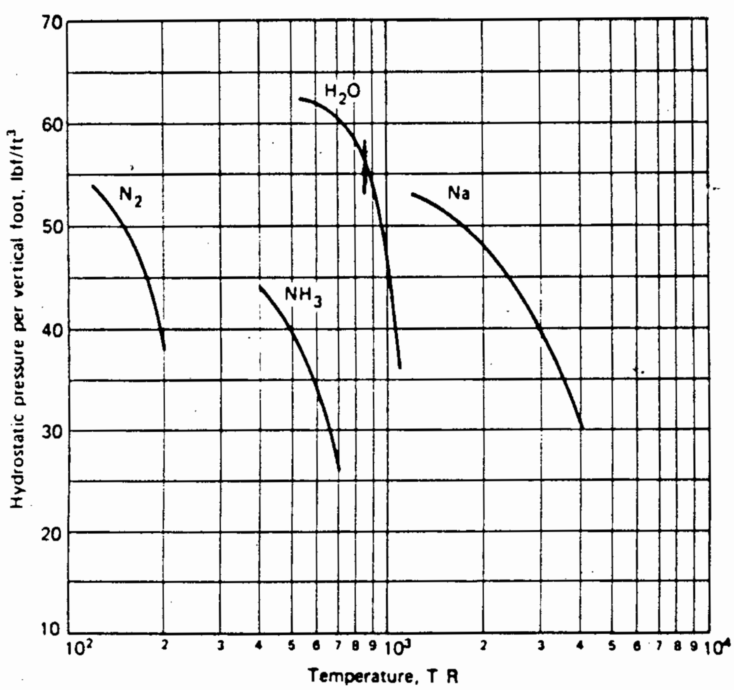


FIG. 7-6 Liquid hydrostatic pressure per unit vertical elevation versus temperature in a standard gravitational field (1 lbf/ft³ = 157.1 N/m³, 1R = 0.5556K).

APPENDIX 3.10

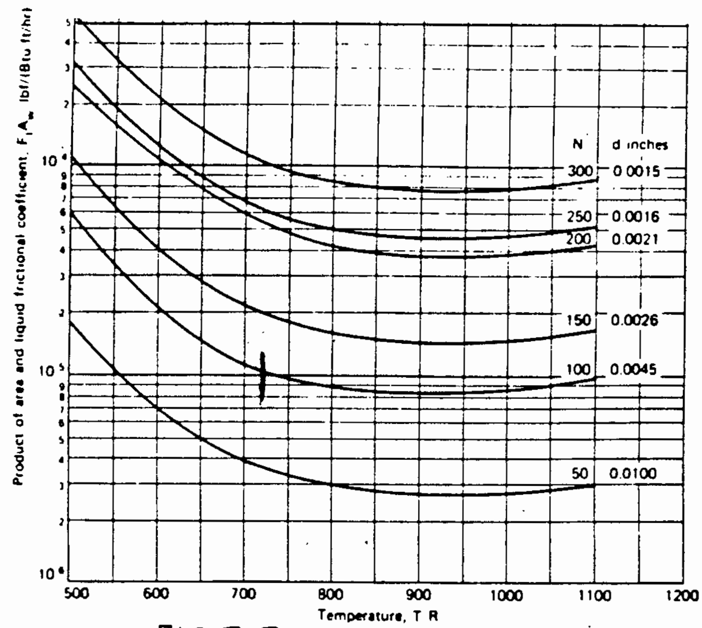
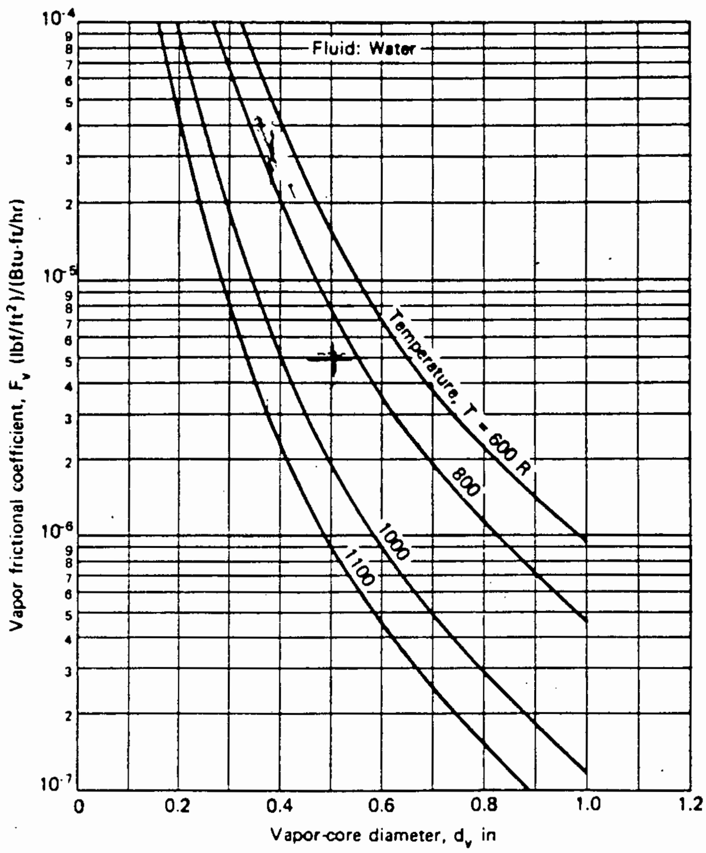


FIG 7-7 (c)



(c)

FIG. 7-8 (*continued*) Vapor frictional coefficient for vapor flow in round tubes [1 (lbf/ft^2)/(Btu-ft/hr) = 536.3 (N/m^2)/W-m, 1 in. = 0.0254 m, 1R = 0.5556K] (*c*) water; (*d*) sodium.

APPENDIX 3.12

HEAT PIPE FLUID INVENTORY

Chapter 2 provides an expression for determining the fluid inventory of the proposed heat pipe, namely:

$$G = (0,8\ell_c \ell_a + 0,8\ell_e)\sqrt[3]{\frac{3Q\mu_1 \rho_1 \pi^2 D^2}{Lg}}$$

			Assumed Values This Project
G	=	Fluid inventory in cc	-
Q	=	Heat input in watts	1000
ℓ_c	=	Length of cooler (cms)	10.0
ℓ_a	=	Length of adeabatic section (cms)	10.0
ℓ_e	=	Length of evaporator (cms)	10.0
μ_1	=	Dynamic viscosity of liquid	0.36
P_1	=	Density of liquid (KJ/cm^3	0.0972
D	=	Diameter of heat pipe (cms)	3.0
L	=	Latent heat of vaporisation (KJ/kg)	2309.0
g	=	Acceleration due to gravity (cm/sec^2)	0.0981

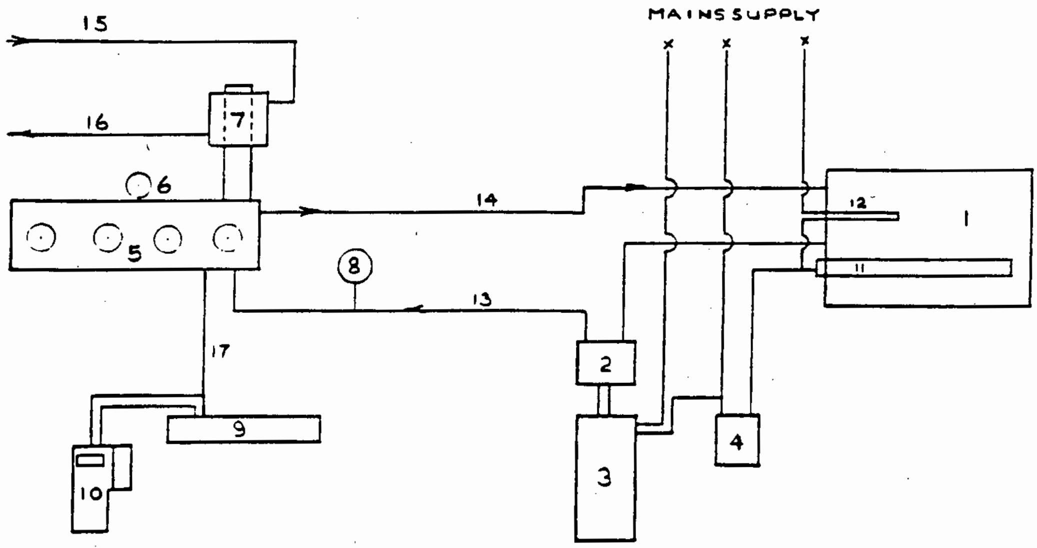
SUBSTITUTING:

$$C = (0.8\times 10 \times 10 + 0.8 \times 10) \times \sqrt[3]{\frac{3 \times 1000 \times 0.36 \times 0.0972 \times TT^2 \times 3^2}{2309 \times 0.0981}}$$

$$= \quad 88 \times 3.451$$

$$= \quad \underline{303.6 \text{ cc}}$$

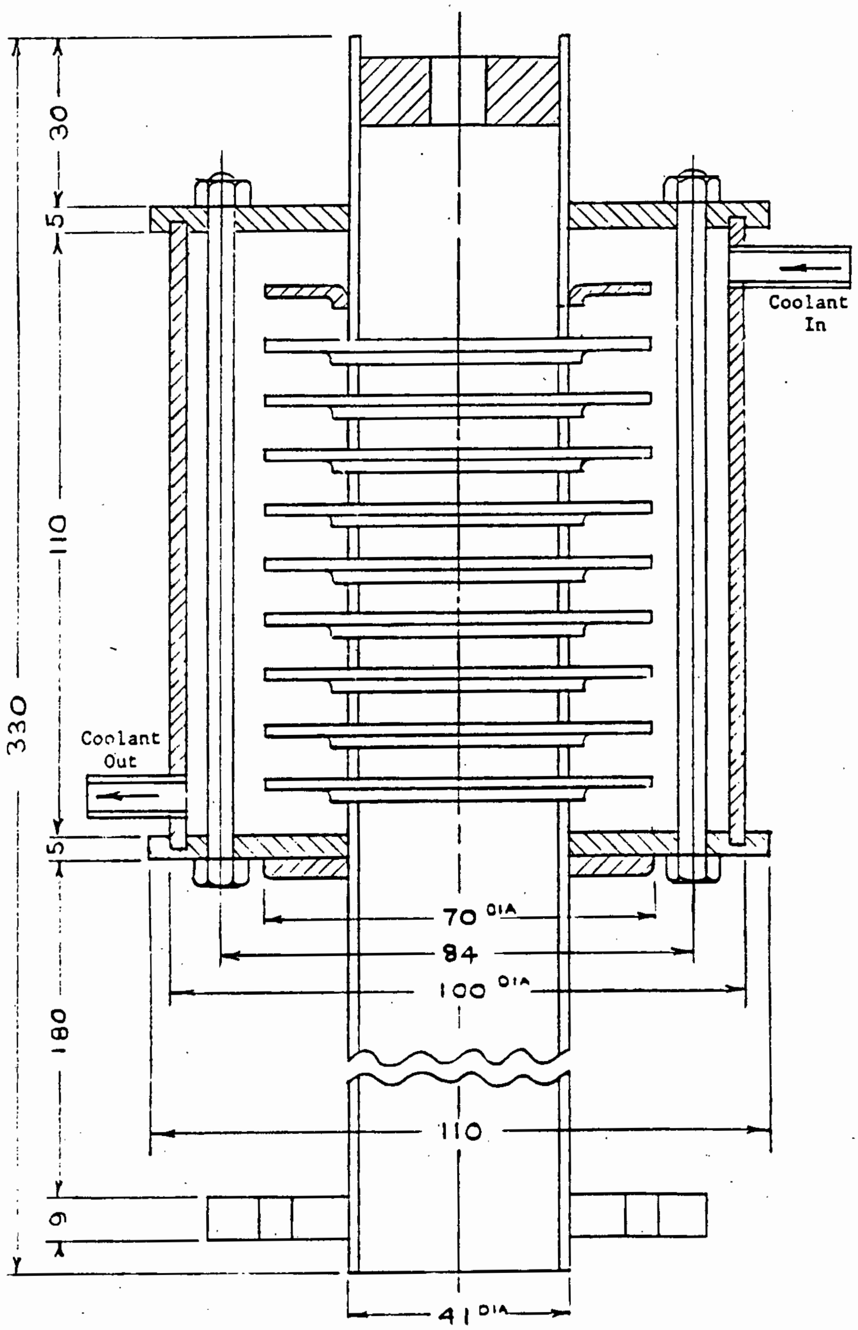
APPENDIX 4.1



APPARATUS SCHEMATIC LAYOUT

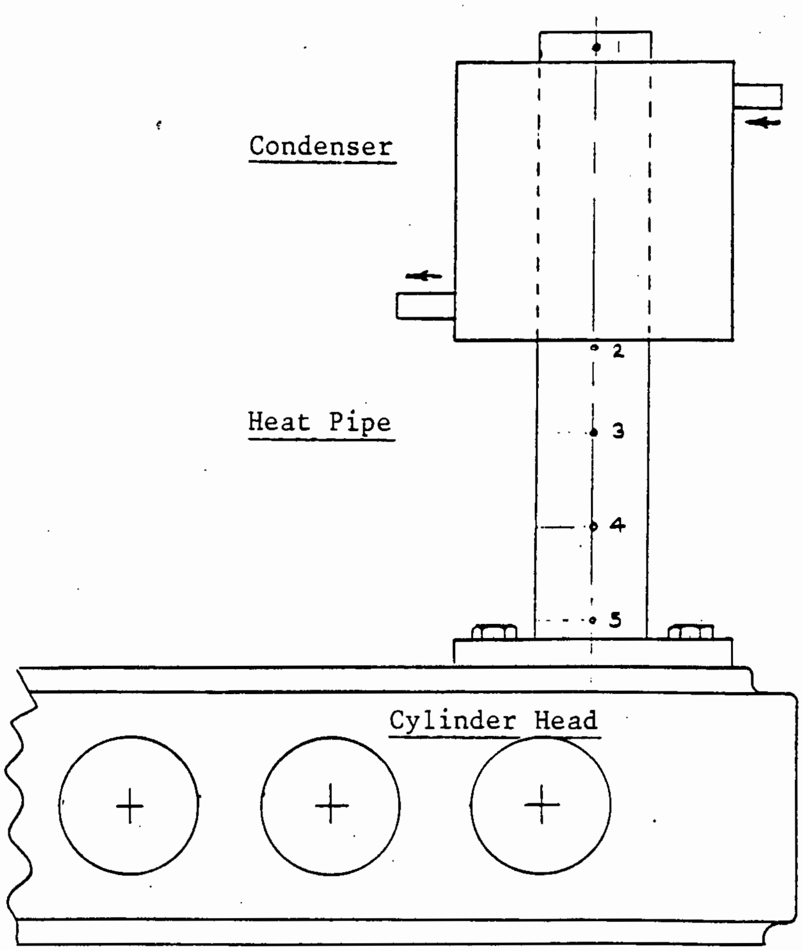
1. Bulk oil tank
2. Oil pump
3. Pump drive motor
4. Digital controller
5. Cylinder head rig
6. System pressure gauge
7. Heat pipe and condenser
8. Oil pressure gauge
9. Thermocouple selector bank
10. Fluke digital multimeter
11. Heating element
12. Thermostat oil temperature
13. Oil delivery pipe
14. Oil return pipe
15. Coolant water inlet
16. Coolant water outlet
17. Thermocouple connection

APPENDIX 4.2



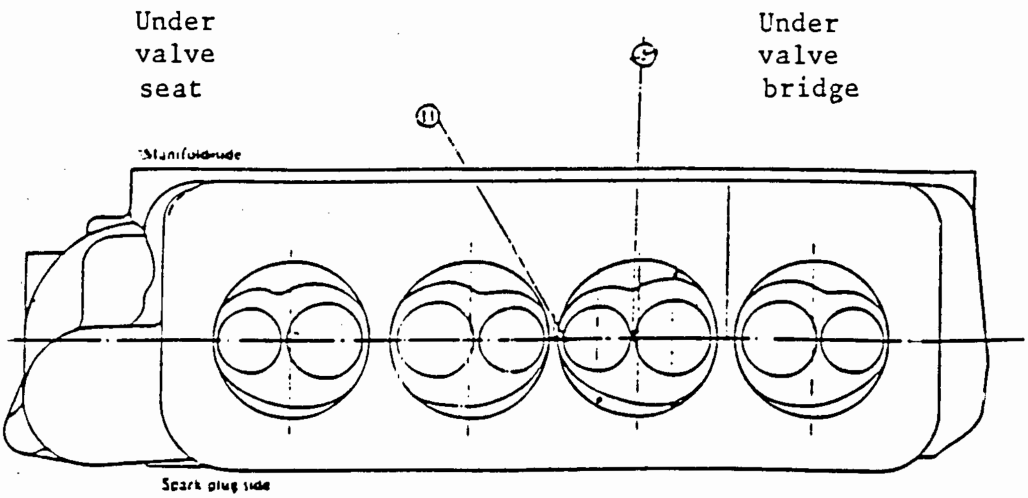
THERMOSYPHON SECTION

APPENDIX 4.3



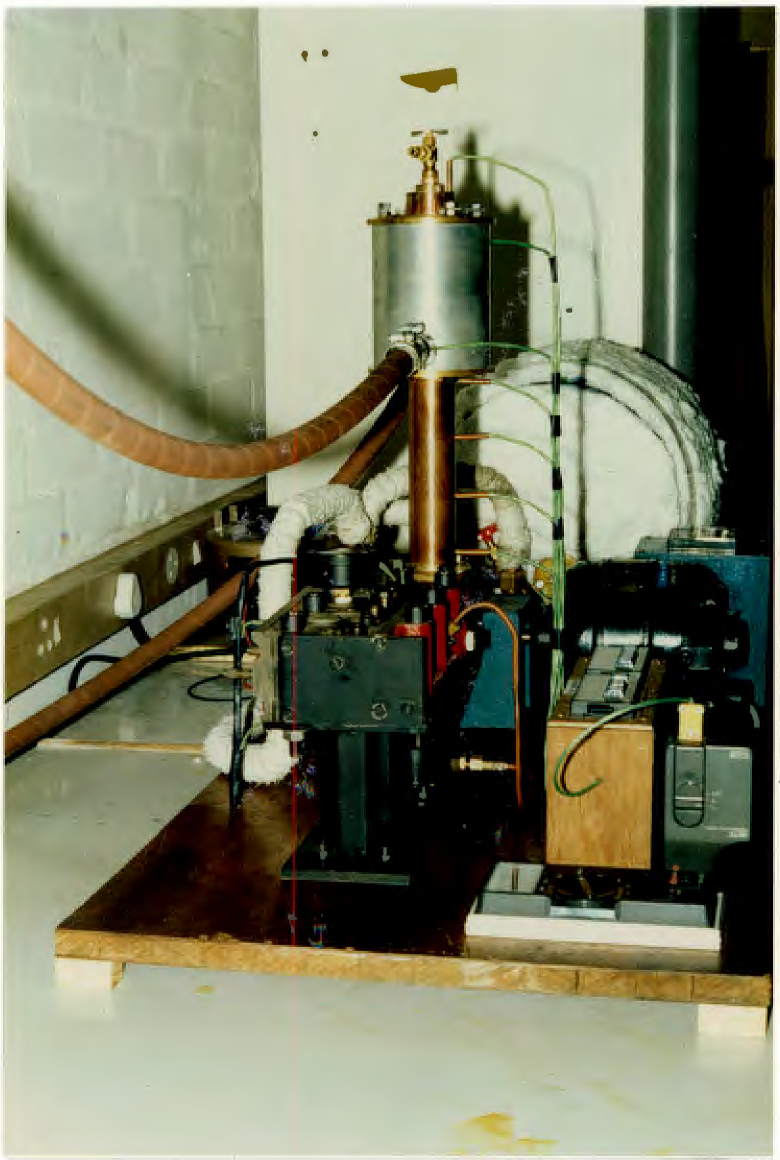
HEAT PIPE THERMOCOUPLE POSITIONS

APPENDIX 4.4



LOCATION OF THERMOCOUPLES IN CYLINDER HEAD

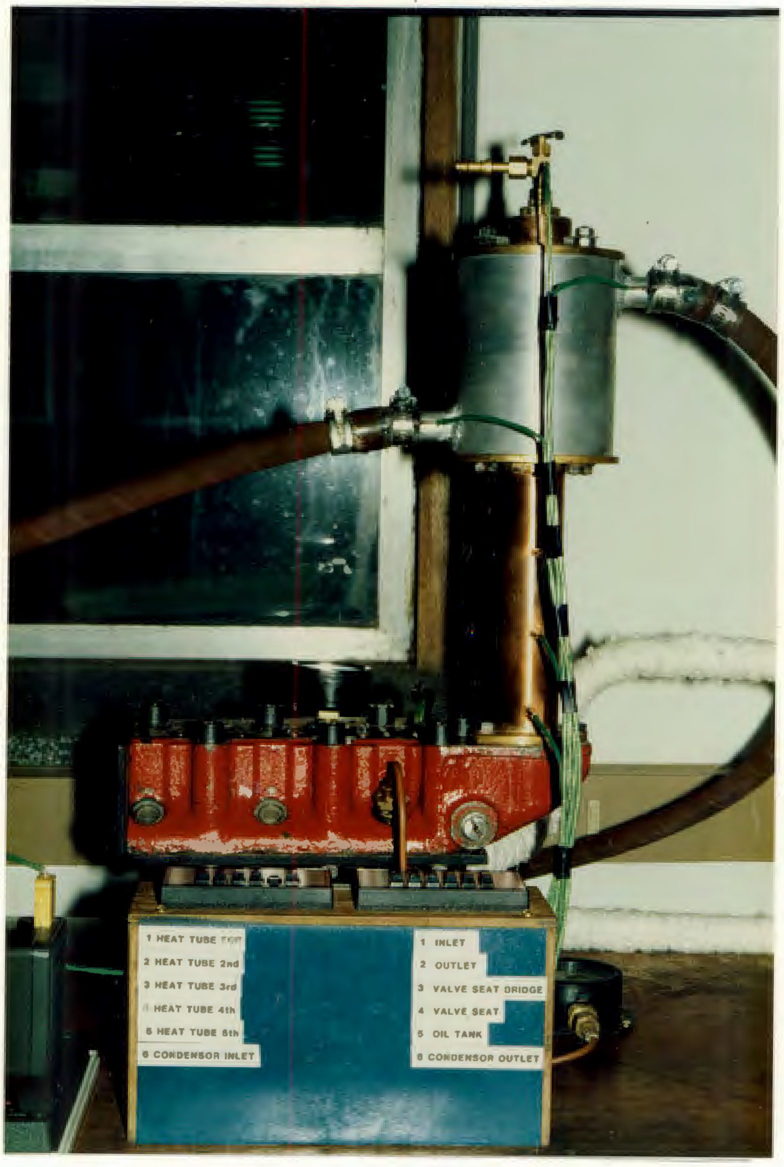
APPENDIX 4.5



HEAT PIPE AND CYLINDER HEAD UNLAGGED

END VIEW

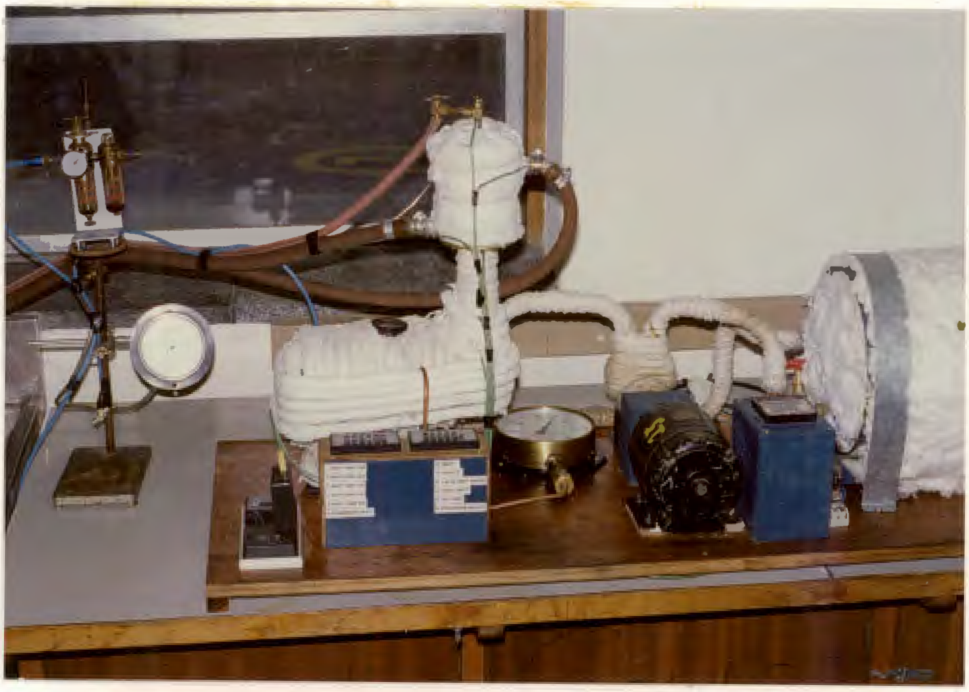
APPENDIX 4.6

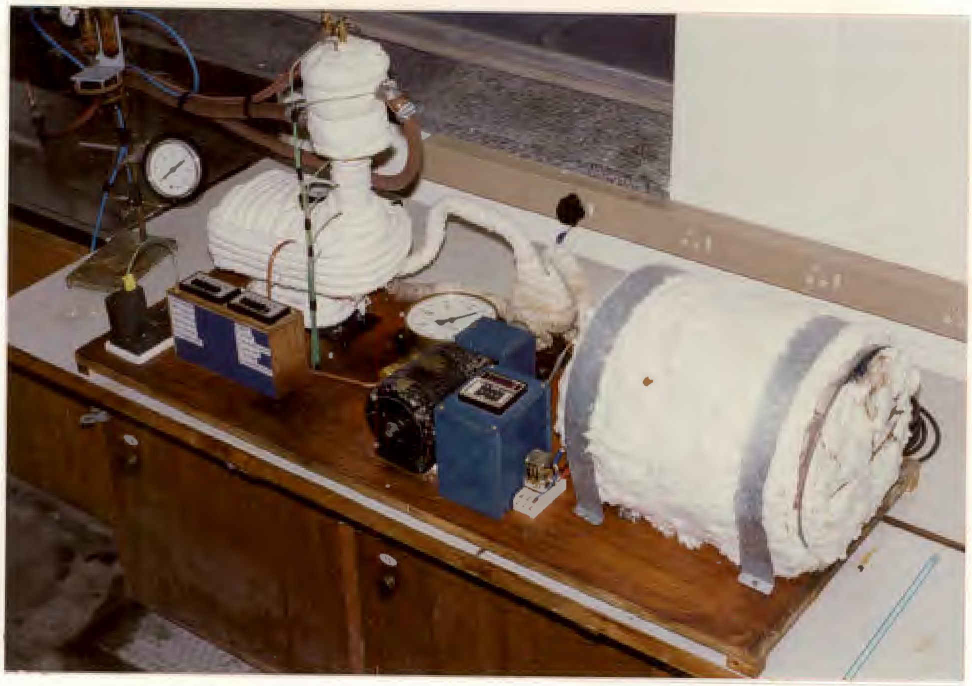


HEAT PIPE AND CYLINDER HEAD UNLAGGED

SIDE VIEW

APPENDIX 4.7



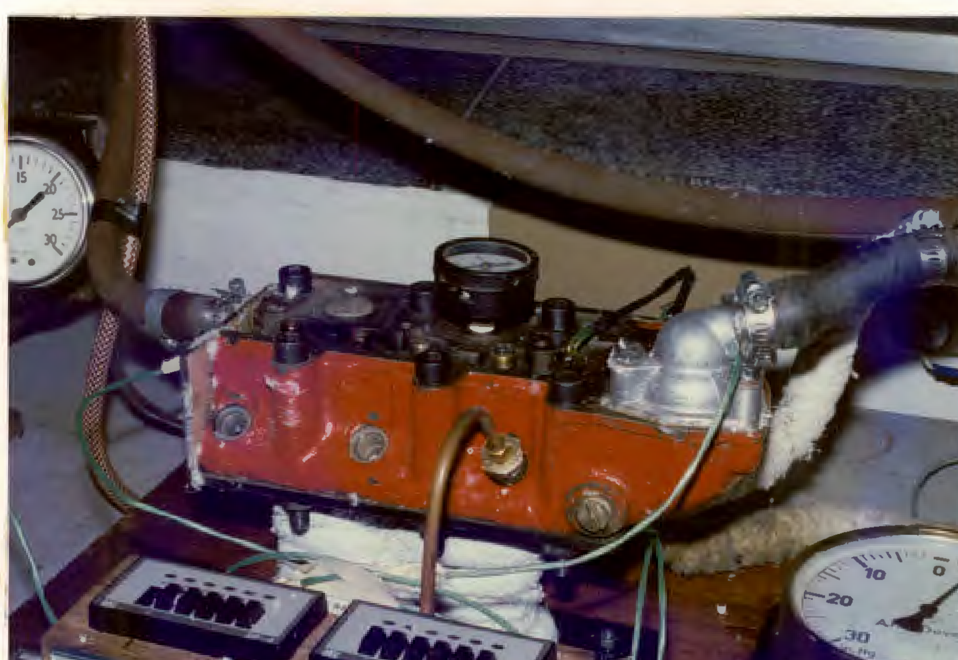


LAGGED CYLINDER HEAD AND HEAT PIPE APPARATUS

APPENDIX 4.9

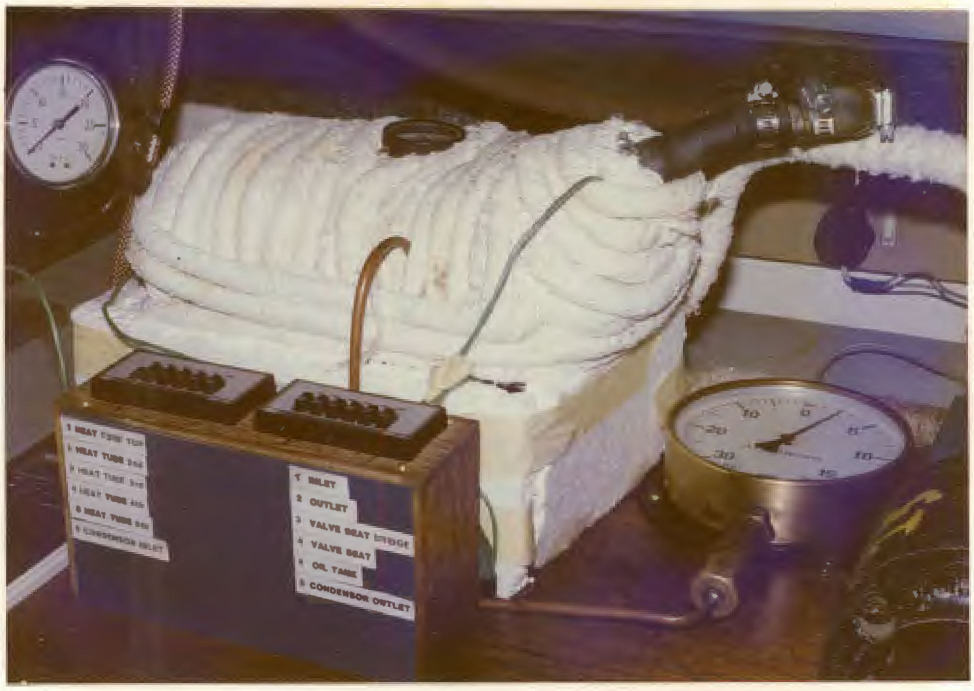
HEAT PIPE AND CONDENSER LAGGED

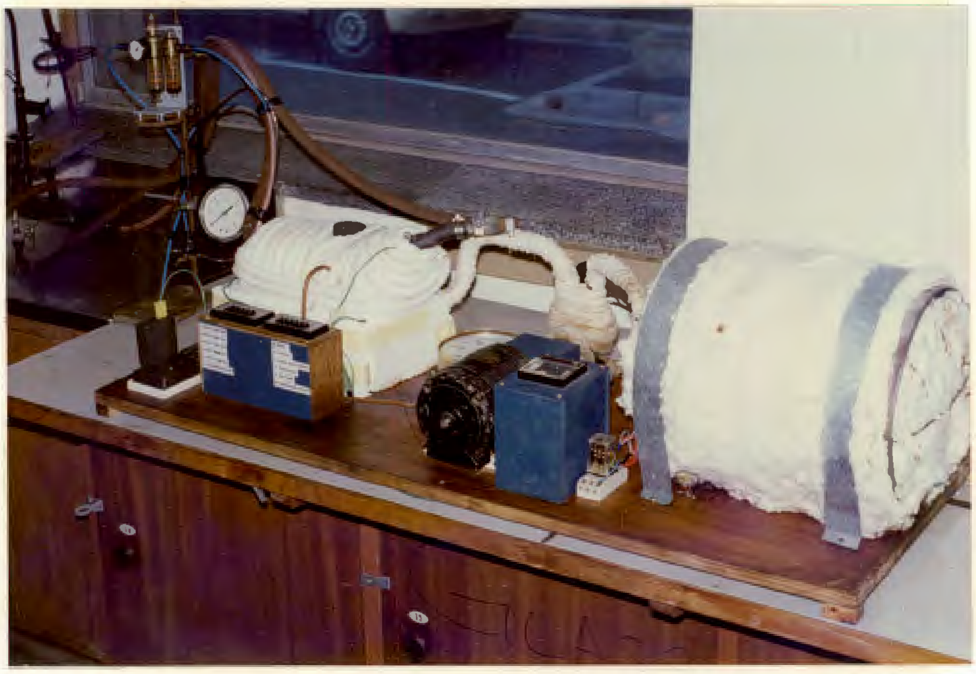
APPENDIX 4.10



CONVENTIONAL SYSTEM APPARATUS UNLAGGED

APPENDIX 4.11





CONVENTIONAL SYSTEM APPARATUS LAGGED

APPENDIX 4.12

CYLINDER HEAD WITH SMALL BORE TUBES

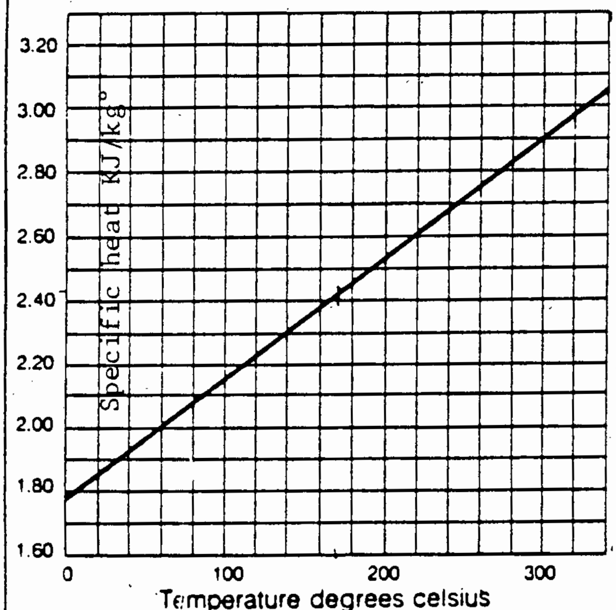
APPENDIX 4.13

FLUID INVENTORY / HEAT PIPE RESPONSE

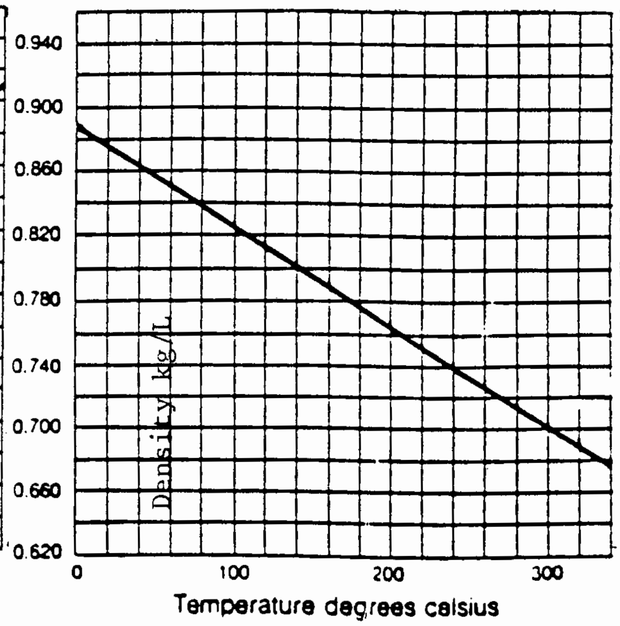
Test No.	Fluid Volume/cc	Response	Sensitivity %
1	250	Low	30
2	400	Fair	60
3	420	Bad	50
4	400	Fair	60
5	520	No	20
6	300	Best	100
7	300	Best	100
8	350	Good	80
9	300	Best	100
10	350	Good	80
11	400	Fair	60
12	350	Good	80
13	400	Fair	60
14	250	Low	30
15	300	Best	100

APPENDIX 4.14

APPARATUS HEATING OIL - SHELL THERMA 'D'



Variation of Specific Heat with Temperature



Variation of Density with Temperature

Typical Characteristics of Shell Thermia Oil D

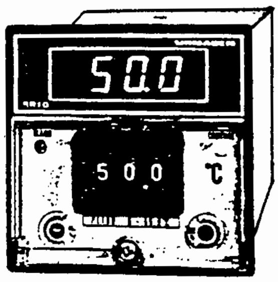
Viscosity, kinematic, cSt, at 20°C		329
ISO 3104	40°C	98
	100°C	11
	200°C	2.1
	300°C	1.0
Viscosity index, ISO 2909		96
Density at 15°C, kg/L, ISO 3675		0.881
ur point, °C, ISO 3016		−9
ash point, closed, °C, ISO 2719		234
Flash point, open, °C, ISO 2592		242
Fire point, °C, ISO 2592		253
Initial boiling point,°C		Above 360
Coefficient of thermal expansion per °C		0.00065
Neutralisation value, mg KOH/g, IP139		0.05

APPENDIX 4.15

APPARATUS HEATING OIL TEMPERATURE

DIGITAL CONTROLLER

Digital Indicating
Series SR10



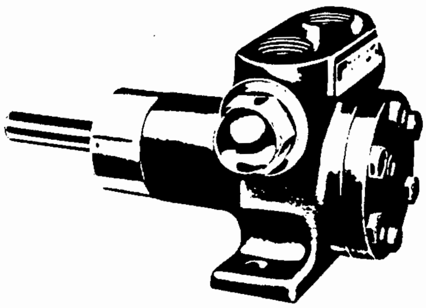
1. SPECIFICATIONS

SR10 Indicating Accuracy . .	±(0.5% + 1-digit) of measuring range
SR11 Deviation Range	±10% of measuring range (vs. setting range)
Thermocouple External Resistance Tolerable Range. .	100 Ω maximum
Cold Junction Temperature Compensation Range	5~45°C (thermocouple input only)
Burn-Out Circuit	Standard (thermocouple input only)
R.T.D. Resistance Tolerable Range	5 Ω maximum/wire
Operating Ambient Temperature & Humidity . .	−10~+50°C, 90% RH maximum
Power Consumption	Approx. 4 VA
Insulation Resistance	500 V DC 20 MΩ minimum between input terminal and power supply terminal 500 V DC 20 MΩ minimum between power supply terminal and earth terminal
Dielectric Strength	One minute at 500 V AC between input terminal and earth terminal One minute at 1000 V AC between power terminal and earth terminal

APPENDIX 4.16

VIKING PUMP

APPARATUS HEATING OIL

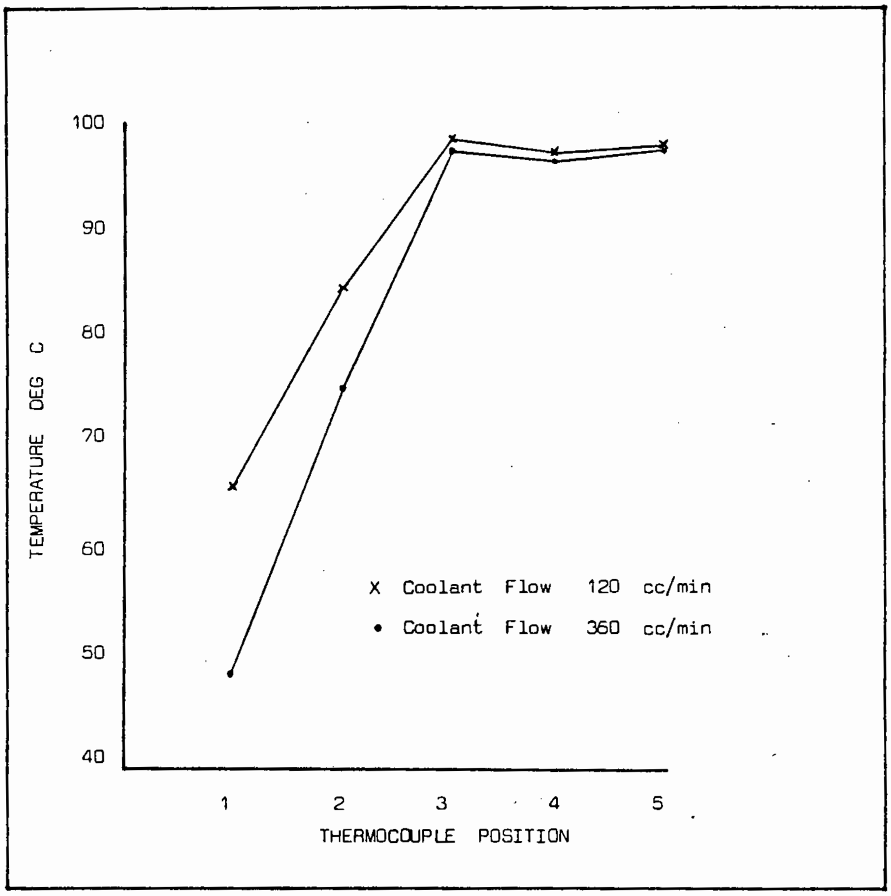


SERIES 32 and 432 "C"-"F"-"FH" sizes—Vertical Ports and valve in casing.

SPECIFICATIONS—SERIES 32 and 432

	Model Numbers Unmounted Pumps: Packed	Model Numbers Unmounted Pumps: Mech. Seal	Suction Discharge Inches	*Nominal Pump Rating: GPM	*Nominal Pump Rating: RPM	‡Maximum Temperature Degrees F.: Packed	‡Maximum Temperature Degrees F.: Mech. Seal	Approx. Ship. Wgt With Valve Pounds
SERIES 32 and 432	C32	C432	¼	½	1800	300	225	5
	F32	F432	½	1½	1800	300	225	6
	FH32	FH432	½	3	1800	300	225	6
	§G32	§G432	1	5	1200	300	225	15
	H32	H432	1	10	1200	300	225	20
	HL32	HL432	1½	20	1200	300	225	25
	J32	....	1¼	20	420	300	......	55
	K32	....	1½	35	420	300	......	65
	KK32	....	2	50	420	300	......	70
	L32	....	2	90	420	300	......	120
	LQ32	....	†2½	90	420	300	......	125
	LL32	....	†3	140	520	300	......	135
	Q32	....	†3	200	350	300	......	335
	M32	....	†4	280	280	300	......	490
	N32	....	†5	450	280	300	......	670

APPENDIX 6.1


100
90
80
70
60
50
40
TEMPERATURE DEG C
1
2
3
4
5
THERMOCOUPLE POSITION
X Coolant Flow 120 cc/min
• Coolant Flow 360 cc/min

APPENDIX 6.2

HEAT PIPE CONCEPT

TIME MINS	TEMP 1 PIPEdegC	TEMP 2 PIPEdegC	TEMP 3 PIPEdegC	TEMP 4 PIPEdegC	TEMP 5 PIPEdegC
0	56.40	84.50	98.20	95.20	93.80
5	59.00	85.40	98.20	95.20	93.90
10	59.40	83.60	98.10	95.20	93.90
15	61.30	85.30	98.20	95.20	94.00
20	61.00	86.20	98.30	95.40	94.20
25	61.20	85.70	98.30	95.40	94.20
30	61.30	86.10	98.40	95.40	94.20
35	62.00	86.30	98.30	95.50	94.20
40	63.30	87.60	98.70	95.60	94.30
45	62.80	86.70	98.30	95.50	94.20
50	62.30	86.80	98.50	95.60	94.20
55	62.70	87.00	98.40	95.50	94.20
60	61.90	86.50	98.50	95.70	94.30
65	60.50	85.30	98.30	95.60	94.40
70	60.80	86.70	98.30	95.50	94.30
75	61.60	86.80	98.10	95.30	94.30
80	63.00	87.70	98.00	95.40	94.30
85	63.80	87.50	98.40	95.50	94.40
90	64.60	88.00	98.50	95.60	94.50
95	63.40	86.90	98.30	95.40	94.40
100	63.00	87.40	98.30	95.50	94.60
105	63.10	87.30	98.20	95.20	94.50
110	64.10	86.80	98.00	94.90	94.40
115	64.10	87.30	98.10	95.00	94.50
120	64.00	86.90	98.00	95.00	94.50
125	59.90	85.10	98.00	95.20	94.50
130	58.80	86.40	97.60	94.80	94.50
135	44.60	.41.60	67.80	82.90	94.80
140	48.80	91.40	111.20	108.10	107.60
145	60.00	93.80	109.00	105.90	105.20
150	65.40	93.20	107.40	104.10	103.60
155	67.20	92.50	105.90	102.90	102.30
160	95.10	90.40	97.90	95.30	94.90
165	65.30	83.00	98.10	95.30	94.60
170	59.50	83.40	98.10	95.30	94.70

APPENDIX 6.2

HEAT PIPE CONCEPT

TIME MINS	TEMP 6 H20 degC	TEMP 7 OIL degC	TEMP 8 OIL degC	TEMP 9 VAL degC	TEMP 10 VAL degC
0	17.60	172.10	170.20	128.00	130.60
5	17.60	170.30	168.40	128.10	130.00
10	17.70	170.80	168.90	127.30	129.50
15	17.80	171.20	169.00	128.20	129.90
20	17.80	172.50	170.30	128.60	130.20
25	17.80	172.90	170.70	128.90	130.50
30	17.90	172.60	170.60	129.00	130.80
35	18.10	171.30	169.40	128.40	130.20
40	18.30	170.30	168.50	127.70	129.50
45	18.00	171.30	169.40	128.20	130.00
50	17.90	171.80	169.80	128.20	130.00
55	18.00	172.40	170.40	128.50	130.00
60	17.80	172.70	170.90	128.60	130.30
65	17.80	170.40	168.60	128.70	129.90
70	17.80	170.80	169.20	128.20	129.60
75	17.90	170.30	168.60	127.80	129.50
80	18.00	171.30	169.50	128.30	129.50
85	18.20	171.50	169.60	128.00	129.20
90	18.10	171.90	170.50	128.20	130.30
95	17.70	169.70	168.50	128.50	130.20
100	17.70	171.40	170.10	128.00	130.10
105	18.10	170.30	168.60	127.90	129.50
110	18.10	170.10	168.30	128.00	128.90
115	18.20	170.40	168.50	128.50	129.10
120	18.10	171.40	169.50	129.10	129.60
125	17.70	172.10	170.10	129.50	129.60
130	17.90	172.60	170.60	130.10	129.80
135	17.90	172.60	170.70	134.90	137.00
140	17.60	170.50	169.00	138.20	138.30
145	17.60	170.30	168.50	135.50	135.50
150	17.70	171.50	169.40	135.50	135.90
155	17.80	172.20	170.10	135.00	134.30
160	18.30	172.90	170.80	130.70	129.60
165	17.30	171.90	169.80	130.10	129.20
170	17.40	171.50	169.40	129.90	129.30

AVERAGE
130.7

APPENDIX 6.2

HEAT PIPE CONCEPT

TIME MINS	TEMP 11 BULKdegC	TEMP 12 H20 degC	T12 - T6 DIFF degC	T7 - T8 DIFF degC	FLOW CC/MIN
0	176.70	41.20	23.60	1.90	224.00
5	175.40	41.40	23.80	1.90	252.00
10	174.20	41.80	24.10	1.90	248.00
15	174.40	43.10	25.30	2.20	228.00
20	175.70	43.00	25.20	2.20	232.00
25	176.20	43.80	26.00	2.20	232.00
30	176.60	44.70	26.80	2.00	228.00
35	175.60	46.00	27.90	1.90	216.00
40	173.80	47.20	28.90	1.80	208.00
45	174.40	45.90	27.90	1.90	224.00
50	174.90	45.70	27.80	2.00	224.00
55	175.70	46.50	28.50	2.00	208.00
60	176.60	44.80	27.00	1.80	248.00
65	175.60	43.80	26.00	1.80	228.00
70	175.10	45.10	27.30	1.60	230.00
75	173.60	45.60	27.70	1.70	220.00
80	174.60	46.50	28.50	1.80	216.00
85	174.80	47.80	29.60	1.90	210.00
90	175.90	48.50	30.40	1.40	200.00
95	176.50	46.80	29.10	1.20	228.00
100	176.30	46.50	28.80	1.30	224.00
105	174.60	46.70	28.60	1.70	220.00
110	174.00	47.60	29.50	1.80	216.00
115	173.90	48.80	30.60	1.90	208.00
120	174.70	46.50	28.40	1.90	212.00
125	175.70	42.80	25.10	2.00	260.00
130	176.30	43.40	25.50	2.00	240.00
135	176.40	35.80	17.90	1.90	232.00
140	175.20	29.00	11.40	1.50	232.00
145	173.80	41.50	23.90	1.80	224.00
150	174.40	43.10	25.40	2.10	220.00
155	175.50	45.20	27.40	2.10	216.00
160	176.40	51.30	33.00	2.10	108.00
165	176.10	36.90	19.60	2.10	400.00
170	175.60	37.00	19.60	2.10	320.00

AVERAGE
230

APPENDIX 6.2

HEAT PIPE CONCEPT

TIME MINS	HEAT/OIL KJ/MIN	HEAT/OIL WATTS	HEAT/H2O KJ/MIN	HEAT/H2O WATTS
0	45.72	762.04	22.10	368.30
5	45.72	762.04	25.07	417.85
10	45.72	762.04	24.98	416.40
15	52.94	882.36	24.11	401.88
20	52.94	882.36	24.44	407.31
25	52.94	882.36	25.21	420.24
30	48.13	802.15	25.54	425.71
35	45.72	762.04	25.19	419.85
40	43.32	721.93	25.13	418.79
45	45.72	762.04	26.12	435.40
50	48.13	802.15	26.03	433.84
55	48.13	802.15	24.78	413.00
60	43.32	721.93	27.99	466.50
65	43.32	721.93	24.78	413.00
70	38.50	641.72	26.25	437.45
75	40.91	681.83	25.47	424.56
80	43.32	721.93	25.73	428.88
85	45.72	762.04	25.98	433.06
90	33.69	561.50	25.42	423.59
95	28.88	481.29	27.73	462.24
100	31.28	521.40	26.97	449.45
105	40.91	681.83	26.30	438.36
110	43.32	721.93	26.64	443.93
115	45.72	762.04	26.61	443.43
120	45.72	762.04	25.17	419.46
125	48.13	802.15	27.28	454.66
130	48.13	802.15	25.58	426.37
135	45.72	762.04	17.36	289.32
140	36.10	601.61	11.06	184.26
145	43.32	721.93	22.38	372.98
150	50.54	842.26	23.36	389.31
155	50.54	842.26	24.74	412.33
160	50.54	842.26	14.90	248.30
165	50.54	842.26	32.77	546.20
170	50.54	842.26	26.22	436.96
	AVERAGE 44.97		AVERAGE 27.58	AVERAGE 460

APPENDIX 6.3

DATA EXTRACT APPEN 6.2

THERMOCOUPLE TEMP degC
THERMOCOUPLE POSITION

	1	2	3	4	5
TEST A					
	77.7	94.2	100.4	98.1	99.0
	71.4	96.9	105.7	103.4	104.3
	98.2	92.2	99.0	96.5	97.2
	48.6	78.6	104.2	101.7	102.5
TEST B					
	72.3	88.0	99.0	96.4	94.6
	56.2	62.1	111.5	108.4	106.4
	58.1	50.0	109.8	107.6	106.6
	58.0	95.9	111.4	108.3	106.2
	97.3	92.5	99.2	96.7	95.0
	77.0	81.9	98.9	96.6	94.8
TEST C					
	58.8	86.4	97.6	94.8	94.5
	44.6	41.6	67.8	82.9	94.8
	48.8	91.4	111.2	108.1	107.6
	60.0	93.8	109.0	105.9	105.2
	65.4	93.2	107.4	104.1	103.6
	67.2	92.5	105.9	102.9	102.3
	95.1	90.4	97.9	95.3	94.9
	65.3	83.0	98.1	95.3	94.6

APPENDIX 6.3

DATA EXTRACT APPEN 6.2

1	VALVE BRIDGE TEMPdegC	HEATPIPE PRESS kPa	COOLANT FLOW CC/MIN	HEAT OUT COOLANT CC/MIN
TEST A				
77.7	133.2	0.0	150.0	27.8
71.4	135.0	35.0	150.0	25.7
98.2	131.4	0.0	150.0	24.2
48.6	138.1	35.0	150.0	16.6
TEST B				
72.3	127.6	0.0	244.0	27.8
56.2	135.4	53.0	240.0	12.8
58.1	135.4	60.0	240.0	21.3
58.0	134.8	50.0	256.0	29.7
97.3	129.3	20.0	176.0	23.5
77.0	130.0	0.0	560.0	31.4
TEST C				
58.8	130.1	0.0	240.0	25.6
44.6	134.9	80.0	232.0	17.4
48.8	138.2	80.0	232.0	11.1
60.0	135.5	80.0	224.0	22.4
65.4	135.5	50.0	220.0	23.4
67.2	135.0	50.0	216.0	24.7
95.1	130.7	0.0	108.0	14.9
65.3	130.1	0.0	400.0	32.8

APPENDIX 6.4

PRECISION COOLING

TIME MINS	TEMP 1 OIL degC	TEMP 2 OIL degC	TEMP 3 H2O degC	TEMP 4 VAL degC	TEMP 5 VAL degC	TEMP 6 VAL degC
0	173.60	174.50	17.30	115.90	118.90	134.80
5.0	172.60	174.50	17.40	118.90	119.50	136.30
10.0	173.10	173.10	17.40	123.50	117.60	135.80
15.0	174.50	175.60	17.40	125.60	118.70	137.20
20.0	172.20	172.90	17.50	126.80	117.20	135.90
25.0	174.50	175.30	16.90	113.00	114.00	131.60
30.0	171.90	173.80	16.60	109.60	114.50	131.10
35.0	174.10	175.50	16.60	111.10	114.50	130.60
40.0	172.60	174.20	16.70	110.90	114.80	131.00
45.0	174.20	175.60	16.80	114.00	114.10	131.40
50.0	172.30	173.80	16.80	112.10	114.40	131.80
55.0	171.80	173.10	16.80	109.10	115.20	131.30
60.0	173.50	175.20	16.90	111.20	115.80	131.30
65.0	172.20	173.30	17.00	110.60	115.20	131.60
70.0	174.20	175.50	17.00	111.40	116.60	132.50
75.0	172.20	173.70	17.00	111.60	115.70	132.00
80.0	172.70	173.60	17.10	111.60	115.40	131.60
85.0	172.00	173.50	17.20	111.30	117.30	133.00
90.0	174.30	175.70	17.30	112.90	116.50	132.80
95.0	173.60	175.20	17.20	117.20	115.20	133.20
100.0	173.00	174.50	17.20	118.00	115.20	132.20
105.0	173.20	174.70	17.40	113.00	117.20	133.40
110.0	173.40	174.90	17.40	114.40	115.40	131.80
115.0	173.40	175.00	16.80	108.00	112.30	128.20
120.0	172.30	173.40	16.20	104.80	110.40	125.70
125.0	173.40	175.00	16.00	105.00	110.30	125.50
130.0	171.90	173.20	16.00	104.30	108.60	124.80
135.0	172.90	174.50	16.20	104.80	109.90	125.90
140.0	171.90	173.00	16.20	104.60	108.40	124.30
145.0	174.20	175.60	17.30	143.60	117.10	138.50
150.0	172.20	173.30	17.70	144.90	114.50	136.60
155.0	173.50	175.10	17.90	126.90	118.20	136.50
160.0	172.00	173.30	18.30	128.00	119.40	137.90
165.0	173.70	174.70	18.60	127.20	117.30	135.20
170.0	172.20	173.60	18.60	128.20	119.00	138.20

AVERAGE
132.33

APPENDIX 6.4

PRECISION COOLING

TIME MINS	TEMP 7 VAL degC	TEMP 8 BULKdegC	TEMP 9 H2O degC	T2 - T1 DIFF degC	T9 - T3 DIFF degC	FLOW CC/MIN
0	121.90	176.50	73.40	0.90	56.10	96.00
5.0	122.70	178.70	75.80	1.90	58.40	96.00
10.0	123.60	176.20	77.20	0.00	59.80	96.00
15.0	124.50	177.00	76.00	1.10	58.60	95.00
20.0	125.70	176.60	78.10	0.70	60.60	92.00
25.0	120.10	177.60	64.20	0.80	47.30	146.00
30.0	118.80	178.70	60.90	1.90	44.30	140.00
35.0	118.60	176.90	65.80	1.40	49.20	126.00
40.0	119.60	178.50	67.20	1.60	50.50	124.00
45.0	119.90	176.80	68.00	1.40	51.20	124.00
50.0	120.30	178.10	69.40	1.50	52.60	120.00
55.0	119.00	176.70	66.50	1.30	49.70	120.00
60.0	120.40	177.40	70.90	1.70	54.00	114.00
65.0	119.40	176.70	70.50	1.10	53.50	112.00
70.0	120.10	176.30	70.50	1.30	53.50	112.00
75.0	120.00	177.70	71.50	1.50	54.50	112.00
80.0	119.80	176.50	73.00	0.90	55.90	110.00
85.0	120.40	177.20	72.40	1.50	55.20	110.00
90.0	120.50	177.40	73.90	1.40	56.60	110.00
95.0	121.20	176.70	69.20	1.60	52.00	108.00
100.0	120.40	175.40	72.30	1.50	55.10	108.00
105.0	121.30	178.60	73.70	1.50	56.30	108.00
110.0	120.00	175.50	73.20	1.50	55.80	102.00
115.0	117.80	178.20	56.60	1.60	39.80	172.00
120.0	115.60	175.90	52.90	1.10	36.70	186.00
125.0	115.30	178.00	49.20	1.60	33.20	220.00
130.0	114.40	176.40	50.60	1.30	34.60	204.00
135.0	115.70	178.20	51.10	1.60	34.90	202.00
140.0	115.30	176.60	50.50	1.10	34.30	192.00
145.0	131.80	177.50	68.50	1.40	51.20	72.00
150.0	132.50	176.00	62.90	1.10	45.20	86.00
155.0	130.70	178.50	92.60	1.60	74.70	70.00
160.0	130.90	177.50	95.60	1.30	77.30	70.00
165.0	130.50	176.50	95.10	1.00	76.50	70.00
170.0	131.70	178.10	89.40	1.40	70.80	68.00
		AVERAGE 177.20				AVERAGE 120.00

APPENDIX 6.4

PRECISION COOLING

TIME MINS	VELOCITY CM/SEC	HEAT/OIL KJ/MIN	HEAT/OIL WATTS	HEAT/H2O KJ/MIN	HEAT/H2O WATTS
0	2.84	23.00	383.33	22.51	375.21
5.0	2.84	48.55	1492.42	23.44	390.59
10.0	2.84	0.00	1528.20	24.00	399.96
15.0	2.81	28.11	1497.53	23.27	387.85
20.0	2.72	17.89	1548.65	23.31	388.42
25.0	4.31	20.44	1208.76	28.87	481.12
30.0	4.14	48.55	1132.10	25.93	432.09
35.0	3.72	35.78	1257.32	25.91	431.89
40.0	3.66	40.89	1290.54	26.18	436.27
45.0	3.66	35.78	1308.43	26.54	442.31
50.0	3.55	38.33	1344.20	26.39	439.75
55.0	3.55	33.22	1270.09	24.93	415.51
60.0	3.37	43.44	1379.98	25.73	428.88
65.0	3.31	28.11	1367.20	25.05	417.46
70.0	3.31	33.22	1367.20	25.05	417.46
75.0	3.31	38.33	1392.76	25.52	425.26
80.0	3.25	23.00	1428.54	25.70	428.39
85.0	3.25	38.33	1410.65	25.38	423.03
90.0	3.25	35.78	1446.42	26.03	433.76
95.0	3.19	40.89	1328.87	23.48	391.26
100.0	3.19	38.33	1408.09	24.88	414.59
105.0	3.19	38.33	1438.76	25.42	423.62
110.0	3.01	38.33	1425.98	23.79	396.53
115.0	5.08	40.89	1017.10	28.62	476.93
120.0	5.5	28.11	937.88	28.53	475.57
125.0	6.5	40.89	848.43	30.53	508.86
130.0	6.03	33.22	884.21	29.51	491.75
135.0	5.97	40.89	891.88	29.47	491.15
140.0	5.67	28.11	876.54	27.53	458.81
145.0	2.13	35.78	1308.43	15.41	256.83
150.0	2.54	28.11	1155.10	16.25	270.82
155.0	2.07	40.89	1908.97	21.86	364.30
160.0	2.07	33.22	1975.42	22.62	376.98
165.0	2.07	25.56	1954.97	22.38	373.08
170.0	2.01	35.78	1809.31	20.12	335.41
		AVERAGE 35.12		AVERAGE 24.86	AVERAGE 414.00

APPENDIX 6.5

PRECISION SYS HEAT TRANSFER COEFF EXP VELOCITIES

TIME	H2O/FLOW	VEL*10^{-2}	h/1 TUBE	h/2 TUBE
mins	cc/min	m/sec	w/m/degC	w/m/degC
mins	cc/min	m/sec	w/m/degC	w/m/degC
0	96	2.8	424.3	848.6
5	96	2.8	424.3	848.6
10	96	2.8	424.3	848.6
15	95	2.8	420.6	841.3
20	92	2.7	410.3	820.6
25	146	4.3	594.2	1188.4
30	140	4.1	574.3	1148.6
35	126	3.7	528.0	1056.0
40	124	3.7	521.4	1042.8
45	124	3.7	521.4	1042.8
50	120	3.5	507.4	1014.8
55	120	3.5	507.4	1014.8
60	114	3.4	486.8	973.6
65	112	3.3	479.4	958.8
70	112	3.3	479.4	958.8
75	112	3.3	479.4	958.8
80	110	3.2	472.8	945.6
85	110	3.2	472.8	945.6
90	110	3.2	472.8	945.6
95	108	3.2	466.2	932.4
100	108	3.2	466.2	932.4
105	108	3.2	466.2	932.4
110	102	3.0	444.9	889.8
115	172	5.1	676.5	1353.0
120	186	5.5	720.6	1441.2
125	220	6.5	824.3	1648.6
130	204	6.0	776.5	1553.0
135	202	6.0	770.7	1541.4
140	192	5.7	739.0	1478.0
145	72	2.1	336.8	673.6
150	86	2.5	388.2	776.4
155	70	2.1	329.4	658.8
160	70	2.1	329.4	658.8
165	70	2.1	329.4	658.8
170	68	2.0	321.3	642.6
	AVERAGE	AVERAGE	AVERAGE	AVERAGE
	120	3.5	570.4	1014.8

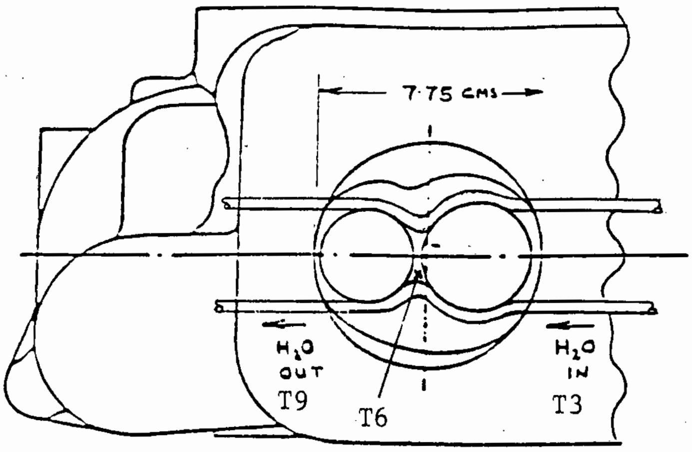
APPENDIX 6.6

PRECISION SYS HEAT TRANSFER COEFF EXP RESULTS

TIME MINS	VALVE TEMP degC	H2O TEMP Tw degC	TEMP DIFF VALVE-Tw degC	APP:6.4 WATTS	EXPMENTAL h W/m²/°C	CALCLATED h W/m²/°C
0	134.8	43.4	91.4	375	1400	848
5	136.3	45.8	90.5	391	1470	848
10	135.8	47.2	88.6	400	1540	848
15	137.2	46.0	91.2	388	1510	841
20	135.9	48.1	87.8	388	1510	821
25	131.6	34.2	97.4	481	1690	1188
30	131.2	30.9	100.2	432	1470	1149
35	130.6	35.8	94.8	432	1560	1056
40	131.0	37.2	93.8	436	1590	1043
45	131.4	38.0	93.4	442	1620	1043
50	131.8	39.4	92.4	440	1630	1015
55	131.3	36.5	94.8	415	1500	1015
60	131.3	40.9	90.4	429	1620	974
65	131.6	40.5	91.1	417	1560	959
70	132.5	40.5	92.0	417	1550	959
75	132.0	41.5	90.5	425	1600	959
80	131.6	43.0	88.6	428	1650	946
85	133.0	42.4	90.6	423	1590	946
90	132.8	49.9	82.9	433	1780	946
95	133.2	39.2	94.0	391	1420	932
100	132.2	42.3	89.9	414	1570	932
105	133.4	43.7	89.7	424	1620	932
110	131.8	43.2	88.6	396	1530	889
115	128.2	26.6	101.6	477	1600	1353
120	125.7	22.9	102.8	475	1630	1440
125	125.5	19.2	106.3	509	1690	1648
130	124.8	20.6	104.2	491	1610	1550
135	125.9	21.1	104.8	491	1610	1540
140	124.3	20.5	103.8	459	1510	1475
145	138.5	38.5	100.0	256	870	670
150	136.6	32.9	103.9	271	890	770
155	136.5	62.6	73.9	364	1680	659
160	137.9	65.6	72.3	377	1780	659
165	135.2	65.1	70.1	373	1820	650
170	138.2	59.4	78.8	335	1450	640

APPENDIX 6.7

This appendix establishes the basis for the values scheduled in Appendix 6.6.



Typical test head section with contoured precision coolant tubes

COOLANT TEMPERATURE Tw

Thermocouples T_3, T_6 and T_9 are shown probed in their relative experimental positions.

T_3 = Temperature of inflowing mains water as coolant
T_6 = Valve bridge metal temperature.
T_9 = Temperature of outflowing coolant

These temperatures were obtained experimentally and recorded in Appendix 6.4.

Tw = Predicted water temperature, 30°C less than the outflowing coolant temperature T_9.

= (T_9 - 30°). The inflowing temperature (T_3) being constant at approximately 17°C.

The heat transfer temperature differential, scheduled in Appendix 6.6 is obtained by the difference between the metal temperature (T_6) and water temperature (Tw). This temperature gradient between metal and coolant.

HEAT TRANSFER FROM OBSERVED READINGS

The heat transfer in watts shown in Appendix 6.6 is derived from the standard equation:-

Q = sp heat x density x temp diff x coolant flow
= heat in watts being absorbed by the coolant water.

The temperature difference in this case being the coolant-out temperature minus the coolant-in temperature, i.e. $T_9 - T_3$. A typical calculation is given for the readings taken at time 10 minutes (refer Appendix 6.4).

T_3 = 17,4 °C, T_9 = 77,2°C, $T_9 - T_3$ = 59,8 °C.
Flow = 96 cc/min.

$$\text{Therefore } Q = \frac{4.186 \times 998.0 \times 59.8 \times 96}{10^6} = 23{,}98 \text{ KJ/min}$$

$$= 400 \text{ watts}$$

The surface area of the coolant tubes:-

Tube length = 7.75 cms, dia = 6 mm

Therefore surface area = 2 x TT (0.6) x 7.75
= 29,2 cm^2.

CALCULATED HEAT TRANSFER COEFFICIENT FROM OBSERVED READINGS
(BASIS FOR ARRIVING AT HEAT TRANSFER COEFFICIENT (h) FOR APPENDIX 6.6)

$$h = \frac{\text{HEAT TRANSFER (WATTS)}}{\text{TUBE AREA X } \{T_6 - (T_9 - 30)\}}$$

For readings at time 10 minutes:-

$$h = \frac{400}{29{,}2x\{135{,}8-77{,}2+30\}}$$

$$= 1540 \text{ watts/m}^2/^\circ C$$

APPENDIX 6.8

CONVENTIONAL COOLING SYSTEM

TIME MINS	TEMP 1 OIL degC	TEMP 2 OIL degC	TEMP 3 H2O degC	TEMP 4 VAL degC	TEMP 5 VAL degC
0	165.20	168.20	71.80	104.90	111.00
5	165.20	168.00	73.90	106.10	111.50
10	165.20	167.90	75.70	107.60	113.70
15	165.30	167.90	78.00	109.00	114.90
20	165.40	167.90	80.10	110.50	116.20
25	165.40	168.00	82.60	112.00	117.90
30	165.50	168.10	84.70	114.90	119.20
35	166.00	168.50	86.30	116.20	120.40
40	166.20	168.70	88.00	117.20	121.70
45	166.80	169.20	89.50	118.70	122.90
50	167.20	169.70	90.80	119.90	124.20
55	167.60	170.10	92.00	118.90	124.40
60	166.50	169.10	92.50	120.90	124.90
65	165.50	168.30	92.90	121.20	125.20
70	164.90	167.10	93.90	120.60	125.00
75	164.20	166.50	93.80	120.30	124.20
80	163.30	165.30	93.80	119.90	123.30
85	163.40	165.50	93.80	120.50	124.00
90	167.90	170.10	96.10	123.90	127.00
95	169.90	172.30	98.20	126.50	129.20
100	170.70	173.10	99.80	126.70	129.60
105	170.50	173.40	99.60	126.40	129.40
110	169.90	172.30	98.60	125.90	129.00
115	169.60	172.00	98.90	126.00	128.90
120	169.60	172.30	99.70	126.70	129.20
125	170.90	173.40	100.00	127.80	130.10
130	170.50	173.00	100.60	127.70	130.10
135	170.40	172.80	101.50	127.80	130.20
140	170.40	172.90	102.30	128.30	130.60
145	170.50	172.90	102.50	128.70	130.90
150	170.50	173.30	100.60	124.60	125.60
155	170.60	173.40	95.60	123.70	125.20
160	170.20	173.00	93.90	123.30	125.00
165	170.10	172.60	95.10	124.60	126.50
170	169.80	172.30	96.60	125.10	127.00

AVERAGE
127.00

APPENDIX 6.8

CONVENTIONAL COOLING SYSTEM

TIME MINS	TEMP 6 BULKdegC	TEMP 7 H2O degC	T3 - T7 DIFF degC	T2 - T1 DIFF degC	FLOW CC/MIN
0	175.90	16.50	55.30	3.00	144
5	175.00	16.70	57.20	2.80	148
10	174.60	16.70	59.00	2.70	136
15	174.70	16.90	61.10	2.60	120
20	174.60	17.10	63.00	2.50	120
25	174.60	17.40	65.20	2.60	116
30	174.50	17.50	67.20	2.60	112
35	175.00	17.70	68.60	2.50	108
40	175.10	17.80	70.20	2.50	104
45	175.80	17.90	71.60	2.40	100
50	176.40	17.80	73.00	2.50	98
55	176.70	17.90	74.10	2.50	102
60	176.20	17.80	74.70	2.60	98
65	174.90	17.80	75.10	2.80	96
70	173.20	17.70	76.20	2.20	96
75	172.50	17.60	76.20	2.30	95
80	171.30	17.50	76.30	2.00	92
85	171.90	17.40	76.40	2.10	90
90	176.60	17.40	78.70	2.20	92
95	178.70	17.60	80.60	2.40	86
100	179.50	17.70	82.10	2.40	94
105	179.80	17.70	81.90	2.90	95
110	178.20	17.70	80.90	2.40	95
115	178.20	17.70	81.20	2.40	94
120	178.70	17.70	82.00	2.70	92
125	179.60	17.80	82.20	2.50	90
130	179.10	17.80	82.80	2.50	90
135	178.80	17.80	83.70	2.40	90
140	179.30	17.90	84.40	2.50	85
145	179.20	17.90	84.60	2.40	87
150	180.00	18.10	82.50	2.80	122
155	179.60	18.10	77.50	2.80	124
160	179.30	17.90	76.00	2.80	102
165	178.70	18.00	77.10	2.50	100
170	178.50	18.00	78.60	2.50	98

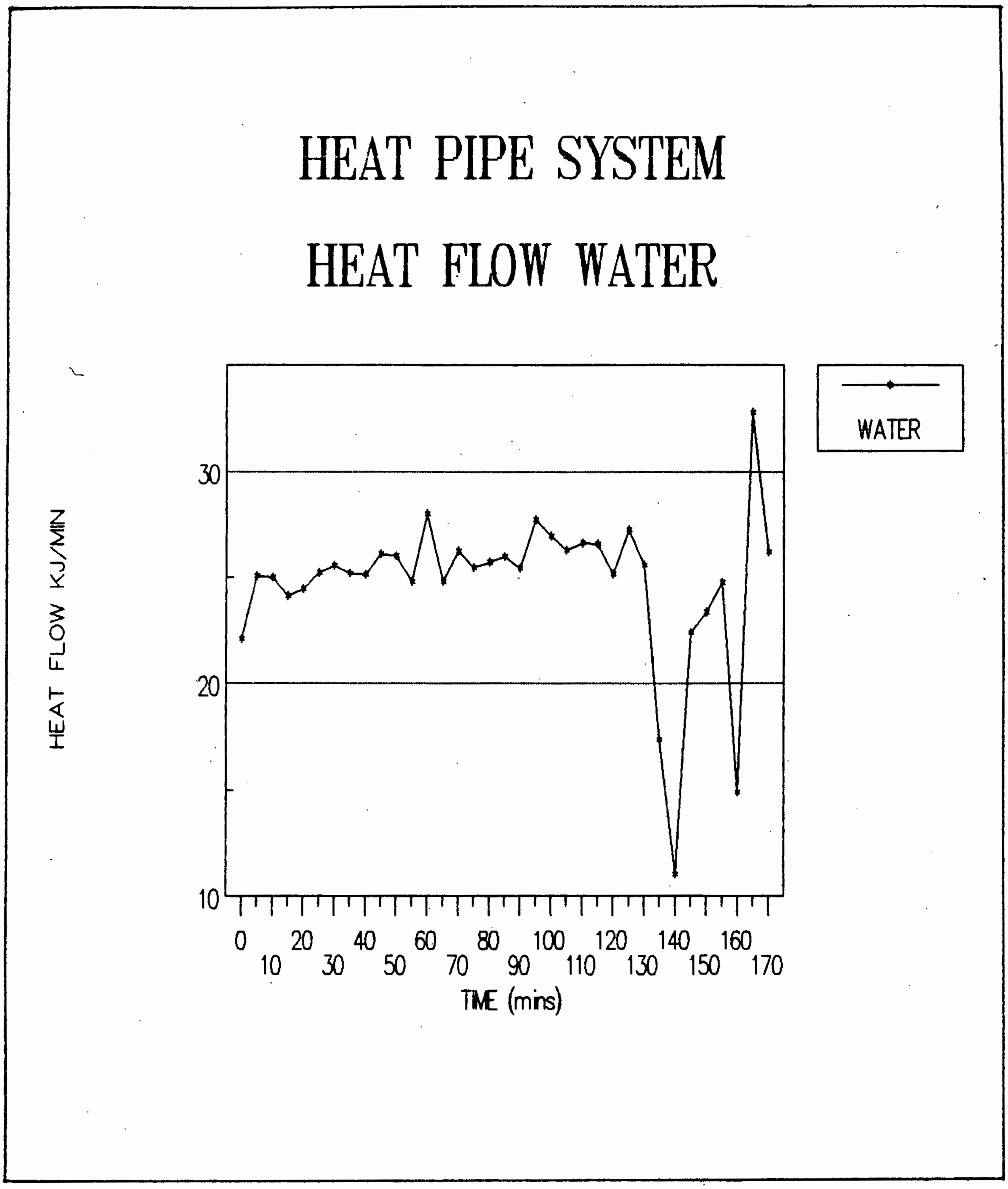
AVERAGE
177.4

APPENDIX 6.8

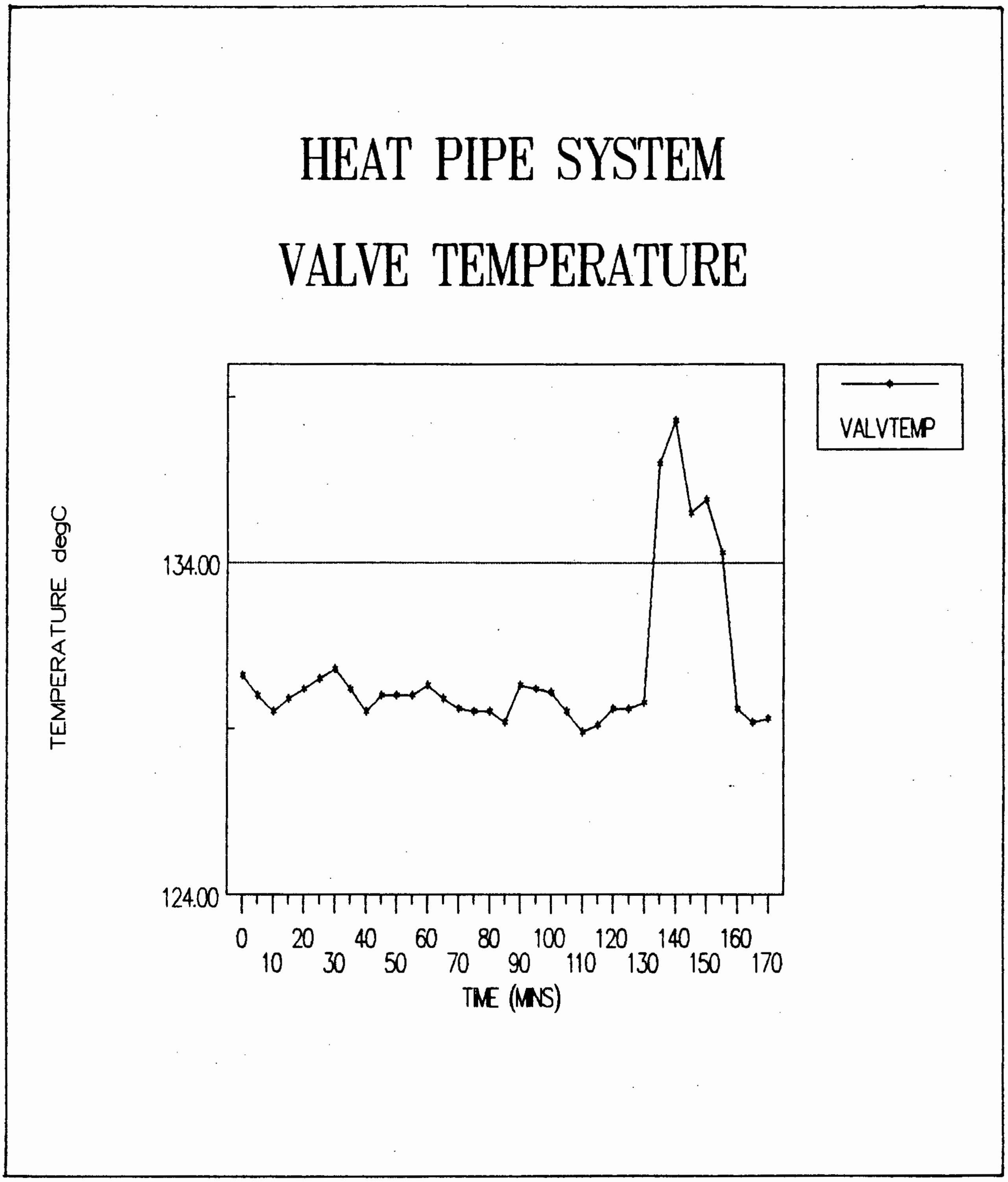
CONVENTIONAL COOLING SYSTEM

TIME MINS	HEAT/OIL KJ/MIN	HEAT/OIL WATTS	HEAT/H20 KJ/MIN	HEAT/H20 WATTS
0	72.19	1203.22	33.29	554.79
5	67.38	1123.01	35.39	589.79
10	64.97	1082.90	33.54	559.02
15	62.57	1042.79	30.65	510.81
20	60.16	1002.69	31.60	526.70
25	62.57	1042.79	31.62	526.92
30	62.57	1042.79	31.46	524.36
35	60.16	1002.69	30.97	516.16
40	60.16	1002.69	30.52	508.64
45	57.75	962.58	29.93	498.83
50	60.16	1002.69	29.90	498.41
55	60.16	1002.69	31.59	526.57
60	62.57	1042.79	30.60	510.02
65	67.38	1123.01	30.14	502.29
70	52.94	882.36	30.58	509.64
75	55.35	922.47	30.26	504.33
80	48.13	802.15	29.34	489.05
85	50.54	842.26	28.74	479.04
90	52.94	882.36	30.27	504.43
95	57.75	962.58	28.98	482.92
100	57.75	962.58	32.26	537.66
105	69.79	1163.12	32.52	542.06
110	57.75	962.58	32.13	535.44
115	57.75	962.58	31.91	531.77
120	64.97	1082.90	31.53	525.58
125	60.16	1002.69	30.92	515.41
130	60.16	1002.69	31.15	519.17
135	57.75	962.58	31.49	524.82
140	60.16	1002.69	29.99	499.81
145	57.75	962.58	30.77	512.78
150	67.38	1123.01	42.07	701.22
155	67.38	1123.01	40.17	669.52
160	67.38	1123.01	32.40	540.07
165	60.16	1002.69	32.23	537.15
170	60.16	1002.69	32.20	536.67
	AVERAGE 59.97		AVERAGE 31.66	AVERAGE 527.00

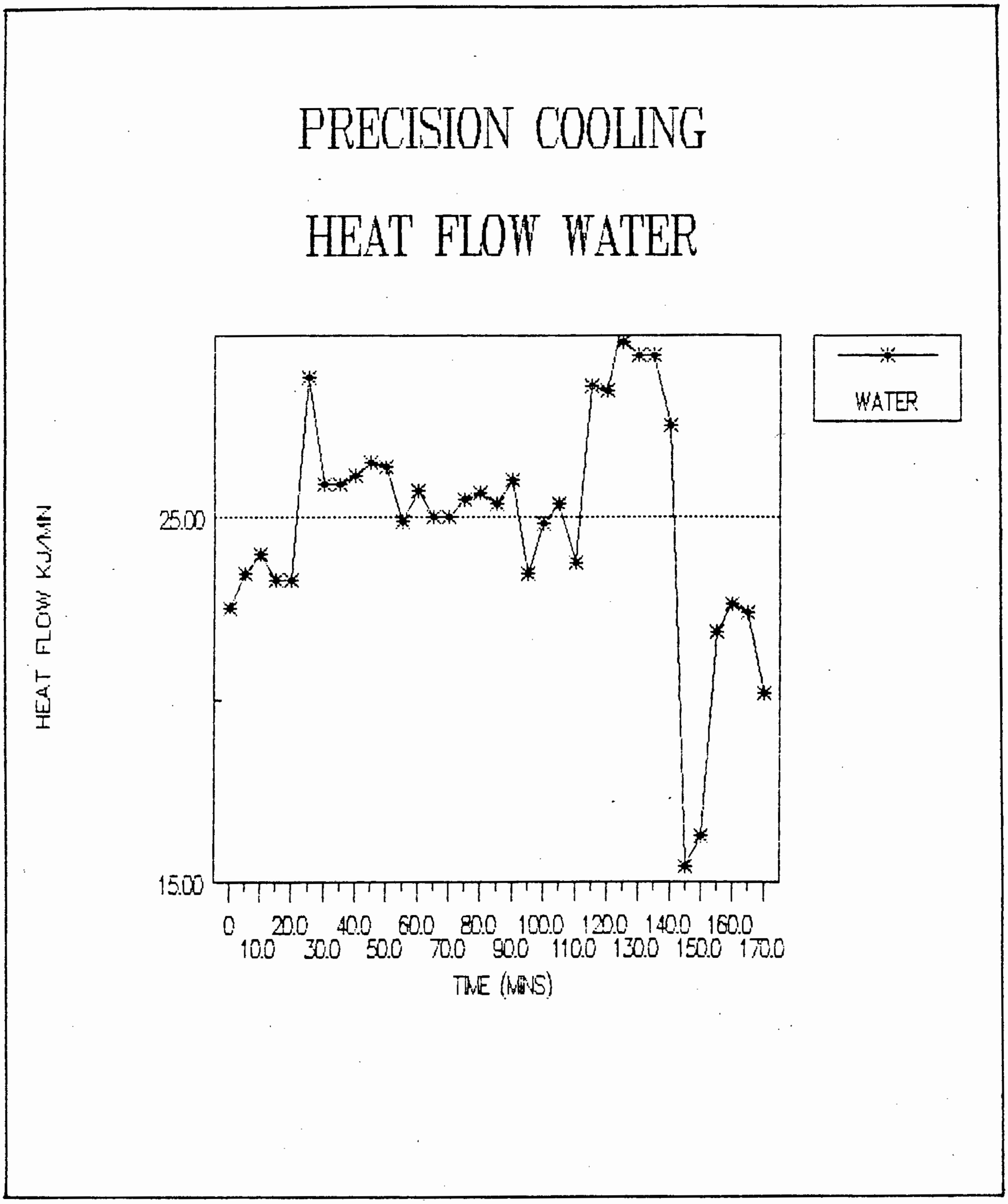
APPENDIX 6.9


HEAT PIPE SYSTEM
HEAT FLOW WATER
WATER
HEAT FLOW KJ/MIN
30
20
10
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
TIME (mins)

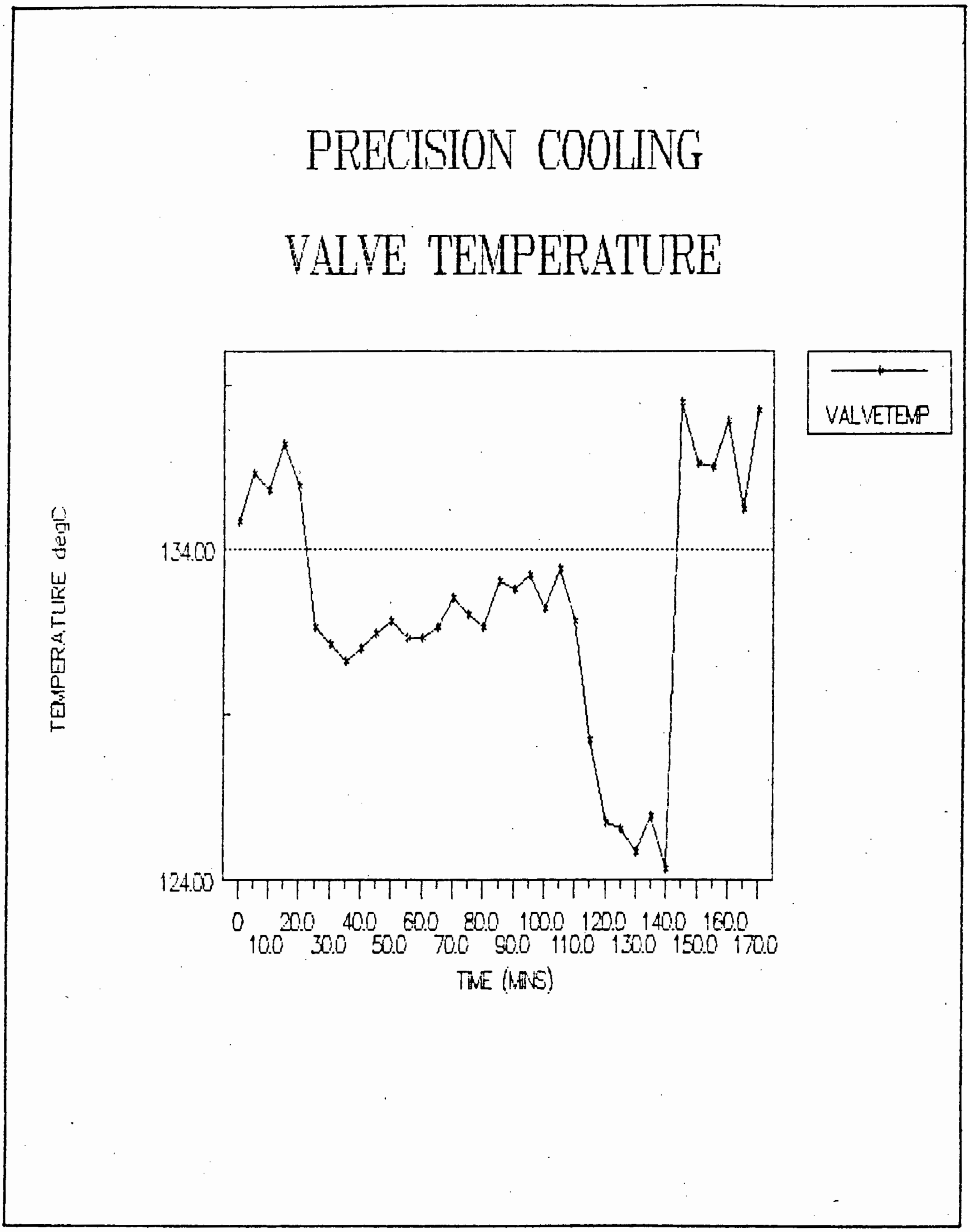
APPENDIX 6.10



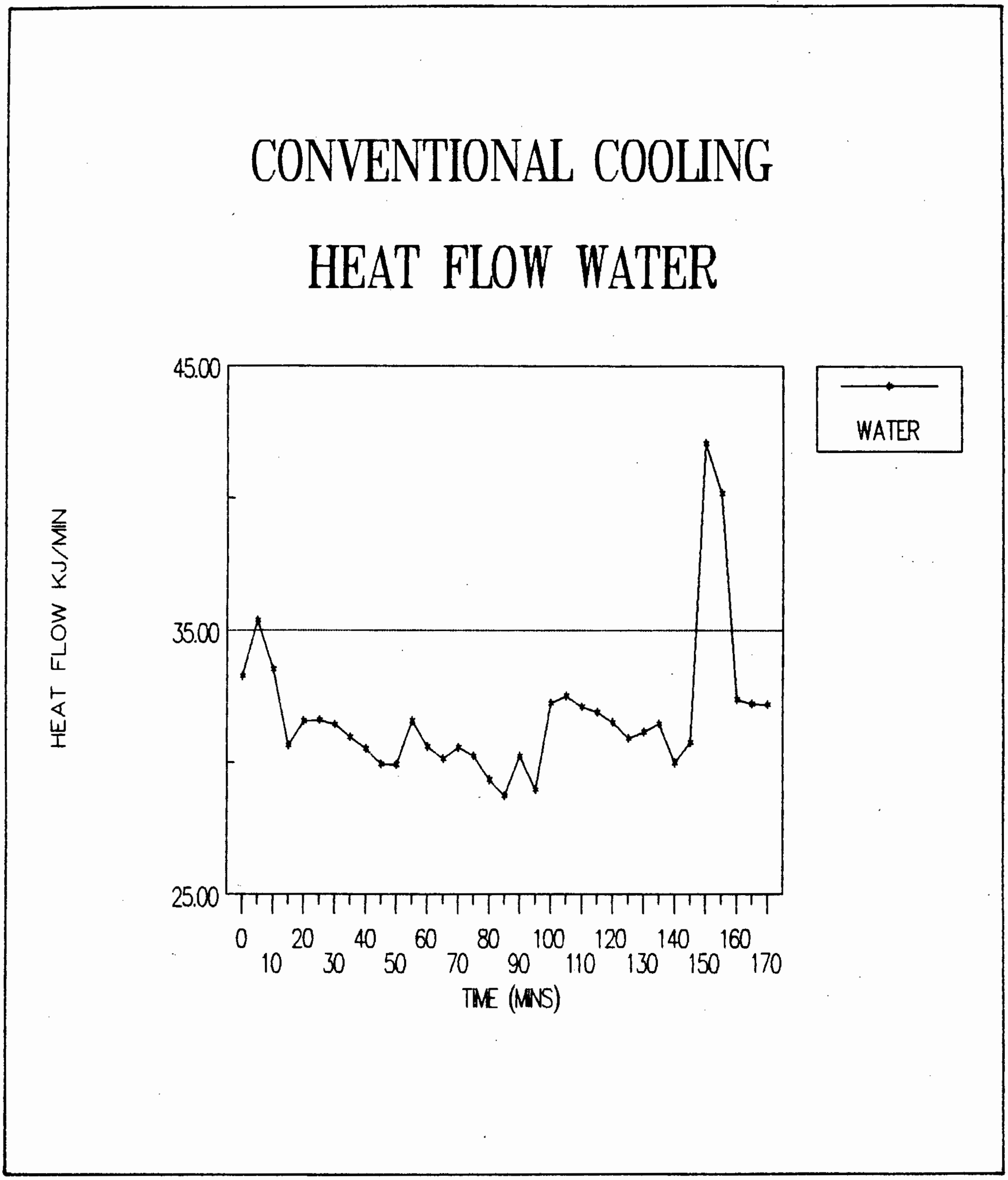
APPENDIX 6.11



APPENDIX 6.12



APPENDIX 6.13



APPENDIX 6.14