Mech. Engg. Res. Bull.
Vol. 10, (1987), p p. 46—53

# Heat-Transfer in Liquid Cooling system of superharged Internal combustion Engine

**Sarkar. M. A. R.*** **Sherbakov A. J.****

**Abstract**

Possobilites of heat-transfer simulation in Liguid cooling system of Internal combustion engine were studied. A model for jacket of Internal combustion Engine was developed. The model is a concentric 'tube in a tube' heat-exchanger. The inner tube is of 50mm diameter and 260mm in length. The outer tube acting as a cooling jacket has its diameter of 74mm, so that the annular gap is of 12mm. Heat-flux on the inner tube snrface was varied from 25-100KW/m². The flow-velocity of cooling water were varied from 0.05-0.5m/sec. The set-up allows to study the basic physical characteristics of heat-transfer in cooling jacket of Internal combustion engine. The correlation obtained from experiments can be used for prediction of heat-transfer coefficient in the cooling jacket of supercharged internal combustion engine.

**Notations :**

$a$ — thermal diffusivity

$c_p$ — specific heat

$h$ — heat transfer coefficient

$h_{fg}$ — enthalpy of evaporation

$k$ — thermal conductivity

$k_f$ — $\dfrac{h_{fg}}{c_p . \Delta t_s}$ criteri of Kutateladje

$L$ — coniventional diameter of: bubbles

$$L = \sqrt{\frac{\sigma}{\rho' \rho''}}$$

$Nu$ — Nusseit number $= \dfrac{h.L}{k}$

$P$ — pressure in the cooling jacket

$P_{bar}$ — barometric pressure

$p_e =$ — peclet number $p_e = \dfrac{VL}{a}$

$p_r$ — prandtl number $P_r = \dfrac{\mu c_p}{K}$

$q$ — heat flux on the inner tube

$q_o$ — heat-flux when surface boiling begins to start

$V$ — speed of steam generation $V = \dfrac{q}{\rho'' h_{fg}}$

$w$ — flow velocity of water in the jacket

$t_w$ — surface temperature

*Dept. of Mechanical Engg., BUET, Dhaka

*Astrakhan Technical Institute of Industries and Economcs, USSR

$t_s$ — saturation temperature

$t_{b1}$ — inlet temperature of water in the test section

$t_{b2}$ — out-let temperature of water in the test section

$\triangle t_x$ = $T_W - T_s$ — excess temperature

$\triangle t_s$ = $T_s - T_b$ — degree of subcooling

$t_b$ = $^1/_2(T_{b1}+T_{b2})$ — bulk temperature of the liquid.

$\rho'$ — density of cooling water.

$\rho''$ — density of vapour

$\sigma$ — surface tension force.

$\nu$ — kinematic viscosity.

$\mu$ — absolute viscosity

## Introduction :

Heat-transfer in the cylinder liner cooling jacket is quite a complex phenomenon. Relatively little is known about the mechanism of heat-transfer in the cooling jacket. In the design and development of internal combustion Engines it is frequently desired to calculate the heat-transfer rate from the cylinder surface to cooling liquid Often, an approximate estimate of time average heat-transfer rate from the cylinder is sufficient, but there are many purpose for which a detailed analysis of heat-transfer in the cooling jacket is essential.

The work of Chirkov, and Stefanovskuu [3] showed that the layer adjacent to the cylinder cooling surface in the cooling jacket exhibited three characteristic types of behaviour, in order of decreasing subccoling at a given load of the Engine.

a. Single phase heat tranefer.
b. partial surface boiling (a region of independent bubble formation)

Existing Formulas for Prediction of Surface Boiling Heat-transfer Co-efficient in the Cooling Jacket of Internal Combustion Engine.

| Author Year | Structure of the Formula | Exponent of Parameter or Criteri | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | Re | Pe or q | K | Pr | $\frac{P}{P_{bar}}$ | $\frac{T}{T}$ | $\frac{Pr}{Pr_w}$ |
| Chirkov A. A 1948 | $Nu = 528.10^{-4} Re^{0.75} Pr^{0.4} (\frac{Gr}{Re})^{0.4} (\frac{\Delta T}{T})^{0.1}$ | 0.75 | 0 | 0 | 0.4 | 0 | 0.1 | 0 |
| Stefanovski B.S. 1962 | $Nu = 0.064\ Re^{0.66} Pr^{0.43} (\frac{Pr}{Pr_w})^{0.24} (\frac{q}{q_o})^{0.46}$ | 0.66 | 0.46 | 0 | 0.43 | 0 | 0 | 0.25 |
| Aliau M. S. 1966 | $Nu = Re^{0.66} Pr^{0.4} (0.27 + 0.000125\ q)$ | 0.66 | 1 | 0 | 0.4 | 0 | 0 | 0 |
| Novennikov A.I. 1973 | $Nu = 16.4\ Pe^{0.5} Pr^{0.43} Re^{0.23} K_f^{0.3} (\frac{P}{P_{bar}})^{0.15} (\frac{\rho''}{\rho'})^{0.5}$ | 0.23 | 0.5 | 0.3 | 0.43 | 0.15 | 0 | 0 |

c. Fully developed surface boiling (a region of bubble coalscence)

Different regimes of heat-transfer in the engine cooling jacket is shown in Fig. 1

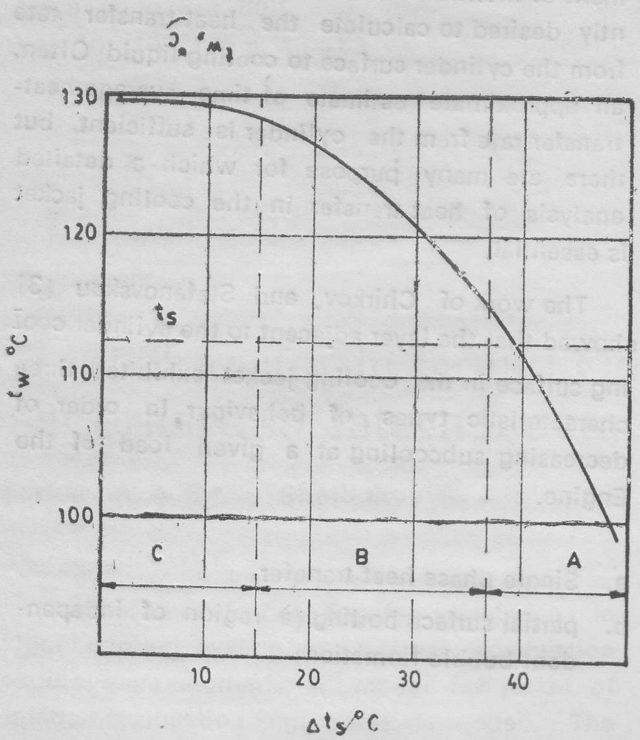



Fig. 1 Effect of boiling poten.ial a cylinder-wall temperature of an automobile engine at atmospheric pressure.

In a natural aspiration engine in case of fractional loading the cylinder wall temperature may not exceed the saturation temperature of the cooling liquid at a predetermined condition. Heat-transfer in this case may occur by natural and forced convection. The cylinder wall temperature increases proportionately with the increase of bulk temperature of the cooling water. Most of the engine designers use formula zonnekena [9] for prediction of forced convention heat-transfer in the cooling jacket.

In modern suoercharged diesel engine the cylinder wall temperature exceeds the saturation temperature of the coolant. Heat-transfer in this case takes place by partial surface boiling. In this regime, once boiling has been initiated, only compartively few nucleation sites are operating so that a portion of the heat will be transfered by normal single phase processes between patches of bubbles. Due to destruction of thermal boundary layer heat-transfer coefficient increases. The rate of change of cylindar-wall temperature decreases with the increase of bulk tamperature of the cooling water as indicated in zone B Fig. 1.

As the cooling temperature further increases the whole surface is covered by bubble sites boiling is fully developed and aingle phase component tends to minimum. Heat transfer in this case is vigorously increased. The rate of change of cylinder wall to temperature is further decreased as shown in zone C Fig. 1. The rate of change of cylinder wall temperature with the decrease of subcooling at a constant load of an engine can be expressed by the foll-owing inequalities,

$$\frac{(dtw)A}{(dtb)A} > \frac{(dtw`B}{(dtb)B} > \frac{(dtw)C}{(dtb)C}$$

Where (dtw)A, (dtw B and (dtw)C are differ-ental changes of wal temperatures in the zone A, B and C and (dtb)A, (dtb)B and (dtb C are differential changes of buik temperature of the liquid in zone, A, B and C respectively. Although fully developed surface boiling heat-transfer is quite helpful for engines as becausa it decreases thermal stress of the cylinder piston assembly as because of increased rate of heat-transfer but in this regime once a vapour bubble froms, its growth rate is very rapid and it quickly may fill tha cooling jacket with vapour. This explosive formation of vapour is often a source of hydrody-namic instability since it is accompanied by a sharp local increase in static pressure which may reduce, stop, even reverse the flow direction of cooling liquid in the jacket. This adverse situa-tion in the cylinder head cooling iacket may occur because of its complex configuration.

The above disscussion concludes that only partial surface boiling heat-rransfer is recommended in the cooling jacket of internal combustion Engine.

Although the process is so important but proper attention has not yet been given on the study of the process. The existing formulae and method are confusing & often contradictory. Authors have different opinions about Physical aspects of surface boiling. Different formula Provide us with results which differ qualitatively and even quantitavely. Some authors have concluded that in surface boiling flow velocity of liquid does not affect practically on heat-transfer coefficient. On the contrary some authors have concluded that flow velocity of liquid affects heat transfer phenomenon. Except that it some suthors have concluded the effect of one parameter on heat-transfer but their dependancy on that parameter are not constant as shown in table (1). This is as because the physical interpretation of the process is not known correctly.

Heat-transfer on surface boiling can be expressed roughly by the functional dependance of the most following pertinent variables.

$$h = f\,(q\ w.\ p.\ \Delta t_s,\ \Delta t_x.\ \rho',\ \rho'',\ \sigma,\ C_p,\ h_{fg},\ k.\ L,\ z)\ [6, 7, 8].$$

Except those parameters heat-transfer coefficients also depend on surface roughess of the cylinder surface and vibration of the heat transfer surface. Complexicity and specifity of the process does not allow to solve problem analtyically. That is why the most reliable method is to use anologus theorms.

The main aim of the work is to get more reliable formula for heat-transfer coefficient what can help us to perform more accurate engineering ealculation. During the choice of pertinent variables we are to avoid the less important variable so as to get sufficient reliable formula which reflects the basic physical characteristics of the process correctly.

**Experimental Facility and Test procedure :**

In order to study the basic physical characteristice of surface boiling phenomenon a model experimental facility was develoded. The schematib diagram of the experimental set-up is shown in Fig 2. The set-up consists of the following main components :

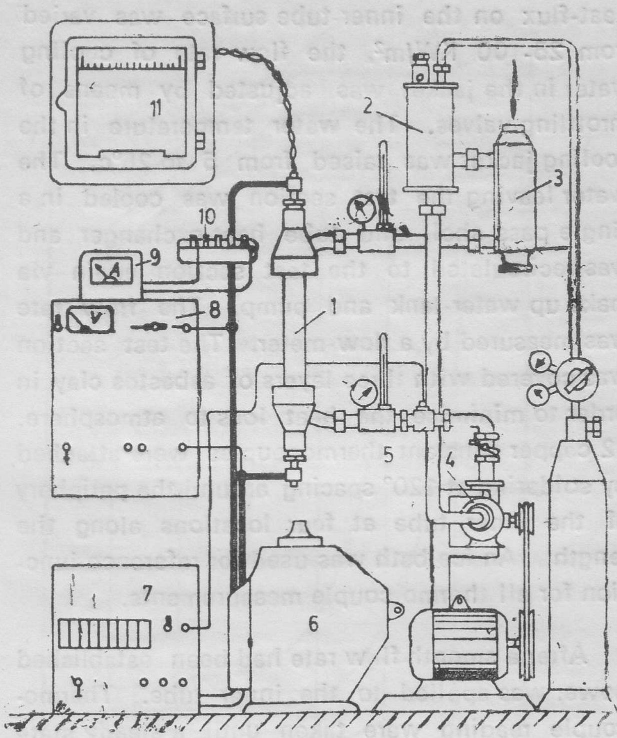



Fig 2 Experimental set up
1-Test section, 2-Make up-water tank, 3-Water cooler, 4-Circulating pump, 5 Flowmeter, 6-Step down transformer, 7-Revolution counter, 8-Precision voltmeter, 9-Precision ammeter, 10-Current transformer, 11-Digital voltmeter.(DVM

1. Test section, 2. Make-up overhead water tank 3. Single pass shell and tube heat exchanger, 4. Circulation pump, 5. Flowmeter 6. Step-down transformer, 7. Revolution counter, 8. Voltmeter, 9. Precision ammeter, 10. Current transformer, 11. Digital Voltmeter.

The test section is a concentric 'tube in a tube' heat exchanger. The inner tube is of 50mm diameter. The outer tube acting as a cooling jecket has its diameter of 74mm, so that the annular gap is of 12mm. The inner tube was heated with an arc welder. The power input was measured with a model wattmeter. Calibration of this meter showed the maximum error on any scale to be less than $\frac{1}{4}$ of 1% full scale value.

Heat-flux on the inner-tube surface was varied from 25-100 $KW/m^2$, the flow rate of cooling water in the jacket was adjusted by means of throttling valves. The water temperature in the cooling jacket was raised from 5 to 25°c. The water leaving the test section was cooled in a single pass shell and tube heat-exchanger and was recirculated to the test section again via make up water-tank and pump. The flow rate was measured by a flow-meter. The test section was covered with three layers of asbestos clay in order to minimise the heat loss to atmosphere. 12 copper constant thermocouples were attached by soldering at 120° spacing around the periphery of the inner tube at four locations along the length. An ice bath was used for reference junction for all thermo-couple measurements.

After a smooth flow rate had been established power was applied to the inner tube. Thermo-couple reading were taken until a steady state was indicated by the absence of a trend. Generally three sets of consecutive reading were sufficient and in average of the last two sets was taken as the result. During the experiments the following parameters were measured.

a) Flow rate of water through test-section.

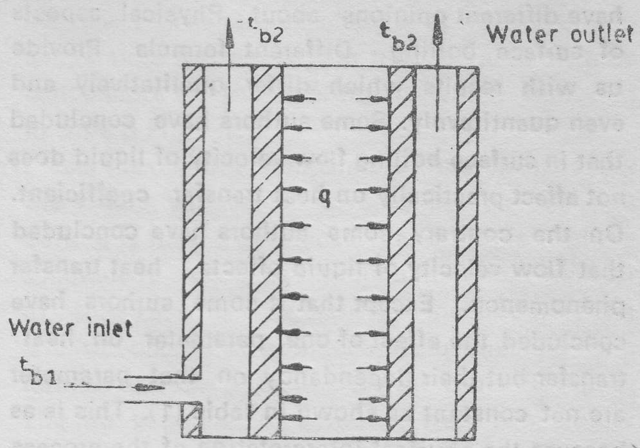
b) Temperature of water at the entrance and exit of the test section.

c) Pressure in the cooling jacket.

d) Heat-flux on the inner tube wall.

## Results and Discussion :

Experiments performed in this work were divided mainly into three different series. In each series of experiments effect of one parameter on heat-transfer coefficients were studied. In each experiments heat-transfer co efficients were written as a function of cooling water temperature. Results obtained from the experiments allow to determine the functional dependance of flow-velocity degree of subcooling heat-flux (q) and pressure in the cooling system on heat-transfer co-efficient. In Fig. 3, and shown




( Test section 1 Fig 2 )

Fig 3 Schematic represention of flow configuration in the test section.

the dependance of the temperatute of the cylinder -wall and heat-transfer co-efficienr on the bulk temparature of the liquid. It is shown that with the increase of bulk temperature, temperature of cylinder-wall and heat transfer co-efficient increases proportionately. But a significicant change in cylinder wall temperature is obsered at a bulk temperature of the cooling water 75°C and above. The rate of chane of cylinder-wall temperature in decreased with the corresponding increase of bulk temperature of the liquid. This is due to the fact that near the cylinder-surface vigorous turbulent a gitation caused by nucleation growth and condensation of bubbles. Thermal resistance of boundary layer is extremely decreased. Simultaneously it is shown in Fig, 4 that flow velocity has no practical influence on surface boiling heat transfer.

In second series of experiments effect of pressure on heat-transfer co efficient were studied at constant heat flux and flow-velocity as shown in Fig. 5. Simulteneously the effect of degree of sub-cooling on heat transfer co-efficient were also determined. The pressure in the system was

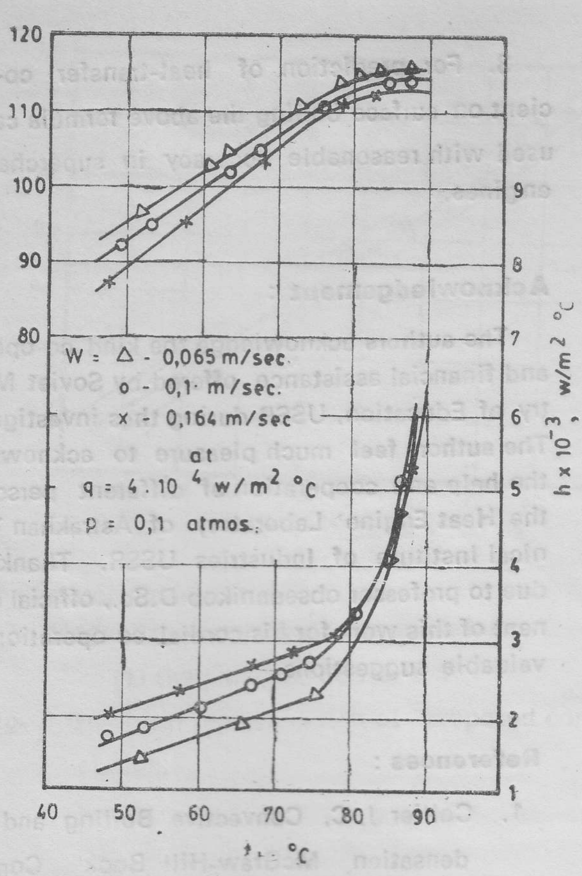



Fig. 4 Effect of flow-Velocity on surface temperature and heat-transfer coefficient at different cooling temperature.

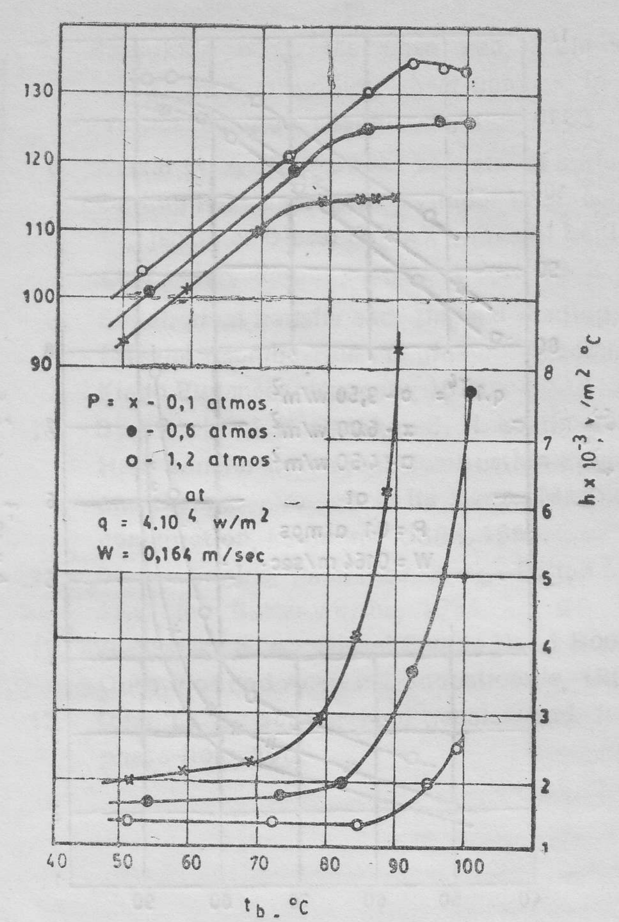



Fig. 5 Effect of pressure on surface temperature and heat transfer coefficient at different cooling temperature.

varied from 0.1 – 1.2 atmosphere and $\Delta t_s = 25-5°_c$.

Results of the experiments shows that

$$h \propto \left(\frac{p}{p_{bar}}\right) 0.5$$

$$h \propto \Delta t_s - 0.35$$

In third series of experiments the effect of heat-flux on heat-transfer co efficient were studied. It is evident from Fig. 6 that with the increase of heat-flux wall temperature increases and surface boiling begins at a lower temperature of the cooling water.

Correlation of the experimental data concluded that the heat-transfer co-efficient can be predicated with the following criterial formula.

$$Nu = 0.03\, P_e^{0.7} \left(1 + 0.72\, K_f^{0.35}\, P_r^{0.3}\right) \left(\frac{p}{p_{bar}}\right)^{0.5} \quad \cdots \cdots \quad 1$$

Further experiments is three four stroke supercharged diesel engine concludes that the formula can be used in predicting heat-transfer coefficients in cooling jacket of engines. None of the formula hither to advanced (table 1) for calculation of transfer co efficients satisfies all requirements for confident extrapolation. But the formula obtained in this work conforms more satisfactorily to the physical requirements and is compatible with a wide range of experimental data.

**Conclusions :**

The conclusions derived from the investigation can be enumerated as follows.

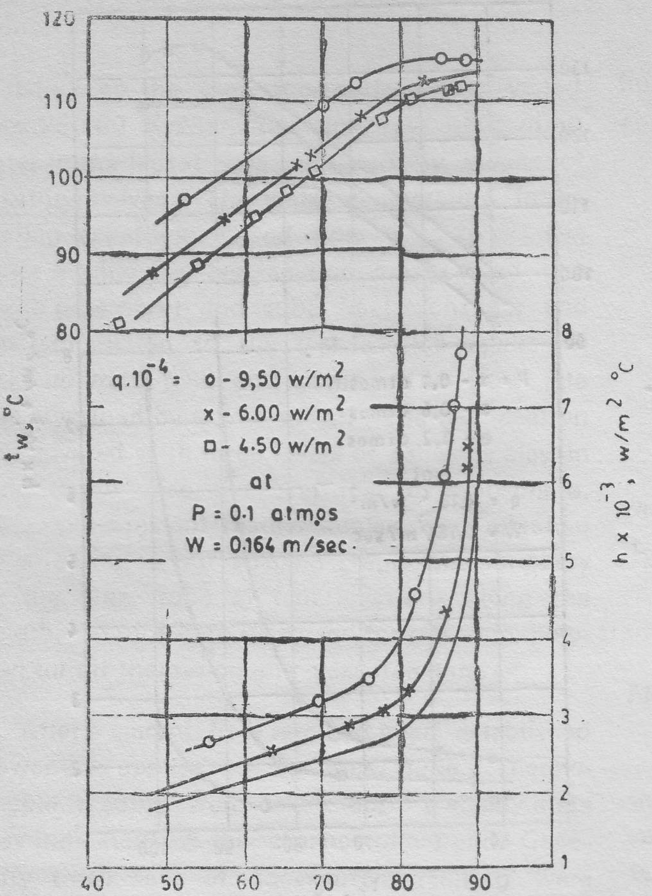



Fig. 6 Effect of Heat-flux on surface temperature and heat-transfer cofficient at different cooling temperature.

1. In surface boiling regime heat-transfer co-efficient may reached upto $600 \frac{k-ca}{m^2 \text{ hr } o_c}$ which is three to four times the heat-transfer co-efficient in forced convection without change in phase.

2. In surface boiling regimes the curves are steep and the wall temperature is practically independent of the fluid velocity which concludes that the agitation caused by bubbles is much more effective than turbulance in forced convection without boiling which defferentiate the present work qualitatively with the other works.

3. For prediction of heat-transfer co-efficient on surface boiling the above formula can be used with reasonable accuracy in superchanged engines.


**Acknowledgement :**

The authors acknowledge the kind co-operation and financial assistance offered by Soviet Ministry of Education, USSR during thes investigation. The authors feel much pleasure to acknowledge the help and cooperation of different persons in the 'Heat Engine' Laboratory of Astrakhan Technical Institute of Industries USSR. Thanks are due to professor obseannikob D.Sc., official opponent of this work for his cordial co-operation and valuable suggestions.


**References :**


1. Collier J. C. Convective Boiling and Condensation, McGraw-Hill Book Company 1972.

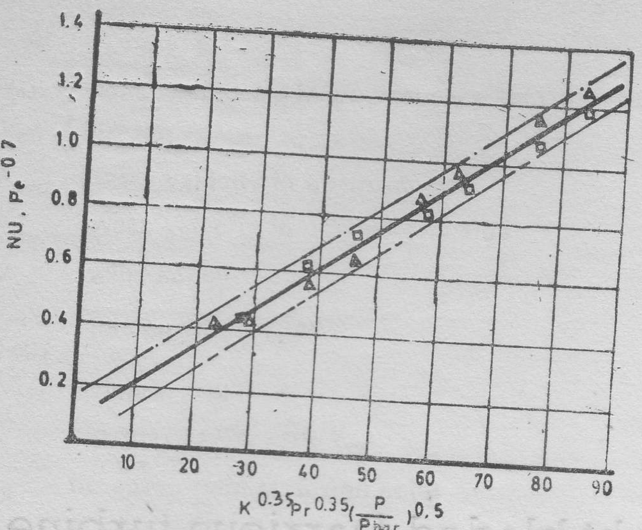
2. Annand, W.J. Heat–transfer in the cylinders of reciprocating Internal Combustion Engine, Proc. Instra. Mech. Engrs., Vol. 177.

3. Stefanobsku B.S. Heat transfer in cooling Jacket of Automobile Engine ETI., 1962.

4. J. Zeder, "New horizons in engine development", SAE Trans., Vol. 6, No. 4, 1952.

5. K Mc. Aulay S chen O Uye hara "Development and evalution of the simulation of the CI Engine, 1965.

6. Aladeeb I. T. Dononod L. D. Experimenta data of sub-cooled boiling heat-transfer in torced-convective and radiation heat-transfer M. Academy of science USSR 1960.




△—Motor experiment

□ Model experiment based on equation

(1) the proposed correlation

Fig. 7 Graphical representation of proposed correlation.


7. Rassokhin N. G. Ma chan Van, Melnicob N. V. Surface boiling heat-transfer in a vertical annulus, cheploenergetica, 1963.

8. Sarkar M. A. R. "Some aspects of surface boiling heat-transfer in cylinder-liner cooling jacket of Diesel Engine, works of Leningrad Polytechuique Institute USSR, 1982.

9. Sitkei, Heat transfer and thermal loading in Internal Combustion, Engines, Akademic Kiado Budapest, Hungary; 1979.

10. Dyachenko N. K. Dashkob S. N. kostin A. K. Heat transfer in Internal Combustion Engine and thermal stresses in its parts. Machine construction, Leningrad USSR, 1969.

11. Diesel Engine Reference Book, Edited by, LRC Lily, Butter worths, 1984.

12. Veshkelsky S. A. Diesel Engine Hand Book, Questions and Answer-L sudostroenie, 1981

13. Tong. L. S. Boiling heat transfer and two phase-flow, 1965.