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EXPERIMENTAL AND NUMERICAL STUDY OF AN INTERNAL COMBUSTION ENGINE COOLANT FLOW DISTRIBUTION

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Original scientific paper

Uniform cooling of all cylinders in an Internal Combustion Engine has been a continual challenge of many engineers and researchers. Different flow rate and velocity of coolant at similar locations of cylinders may cause some regions to be overcooled whereas some to be undercooled. In this study more uniform cooling engine is desired. To do this, at first the governing equations of the flow were solved 1D and 3D to obtain the velocity, pressure and temperature at various points of an existing engine numerically. Then Particle Image Velocimetry (PIV) method on a transparent cylinder head made of Plexiglas was used to validate the numerical simulations. After validation, in order to reach intelligent cooling, some strategies such as small modifications in the engine coolant inlet and outlet along with its flow rates were applied that resulted in more uniform cooling which in turn prevents any over and under cooling.

Keywords: *internal combustion engine; cooling; flow distribution; PIV (Particle Image Velocimetry)*

Eksperimentalna i numerička analiza raspodjele toka rashladnog sredstva u motoru s unutarnjim izgaranjem

Izvorni znanstveni članak

Ujednačeno hlađenje svih cilindara u motoru s unutarnjim izgaranjem predstavlja trajni izazov za mnoge inženjere i istraživače. Različiti tok i brzina strujanja rashladnog sredstva na sličnim mjestima u cilindrima može rezultirati prevelikim ili preslabim hlađenjem nekih mjesta. U ovom se radu želi dobiti motor s ujednačenijim hlađenjem. U tu su svrhu najprije riješene jednadžbe protoka 1D i 3D kako bi se dobili numerički podaci o brzini, tlaku i temperaturi na različitim mjestima postojećeg motora. Tada se primijenila metoda mjerenja brzine fotogramom čestica (Particle Image Velocimetry - PIV) kako bi se na providnom poklopcu glave cilindra izrađenom od pleksiglasa provjeravale numeričke simulacije. Nakon provjere, u svrhu postizanja odgovarajućeg hlađenja, primijenile su se neke strategije kao na primjer male modifikacije na ulazu i izlazu rashladnog sredstva u motor kao i brzine protoka sredstva, a to je rezultiralo ujednačenijim hlađenjem, dakle onemogućavanjem prevelikog ili premalog hlađenja.

Ključne riječi: *motor s unutarnjim izgaranjem; hlađenje; raspodjela toka; mjerenje brzine fotogramom čestica (PIV)*

1 Introduction

The heat released in a combustion chamber of an engine is divided into three main parts. Only about one third of the input energy is converted into useful output power and the rest is wasted somehow by means of exhaust gases and cooling system. The main goal of the cooling system is to keep the engine components at proper temperature. Although the heat rejected to the coolant of an ICE varied with the type, load, and speed of an engine, in general, it is about 17 to 26 % of the input energy in a SI engine and 16 to 35 % in a CI engine [1].

Some regions of the cylinder head such as exhaust valves and valves bridges may experience heat fluxes as high as 10 MW/m^2 during the combustion period [1]. So these regions of the water jacket walls can be threatened by rising temperature which causes some mechanical failure such as deformation and distortion while the overall coolant temperature may remain in the safe range. Different velocities in similar locations in the water jacket may cause under or over cooling. To avoid these problems uniform coolant velocity should be provided at similar locations of water jacket. The concept of providing minimum coolant flow for achievement of optimized temperature distribution is called precision cooling. Ernest [2] seems to be pioneer of this concept. He was focused on a V8 Ford's engine. The main goal of this study was to replace a cast iron cylinder head with aluminium one which results in about 10 % weight reduction. Preide and Anderton [3] studied a heavy duty 3,9 litter diesel engine cylinder head cooling passages. They replaced the original coolant passage with smaller sizes around the valve guides and seats and also injector boss. Finlay et al. [4] applied higher coolant velocities in the thermally critical areas which led to a significant reduction in the bulk coolant flow rate. Arcoumanis et al. [5] used Laser-Doppler Velocimetry technique to characterize local velocities in a transparent model of a production DI diesel engine. They used the experimental results for the computational fluid dynamic analysis validation for further engine water jacket coolant flow investigation. The results were obtained from a transparent model of a Ford 2.5L DI engine. This provided very good insight into the convective cooling capability of the coolant flow especially in critical areas such as the passage between inlet and exhaust ports. Norris et al. [6] studied the thermo-fluid behaviour of the cylinder head cooling analytically and experimentally. His study included a detailed cooling jacket temperature measurements, and finite element heat transfer analysis to identify the regions with pure convection, nucleate boiling, and film boiling. A near zero velocity region, stated as "stagnation", was observed both on the up and downstream sides of the injector sleeve and also in between any two cylinders. More attention has to be focused on these high heat flux regions where adequate cooling is imperative to prevent local hot spots. This could adversely affect the fatigue life of the cylinder head. Temperature measurements of these regions showed that in the critical regions the heat transfer coefficient is higher than the calculated one when pure convection was considered. This indicates that nucleate boiling is the dominant mode of heat transfer in these areas. Kobayashi et al. [7] proposed a dual circuit cooling to reach higher compression ratio. In their work a separate cooling passage was considered for each cylinder block and head.

They found that lowering the cylinder head coolant temperature is more effective in knock reduction than lowering the cylinder block temperature. They also concluded that there was a great possibility of raising compression ratio by lowering the cylinder head coolant temperature while maintaining the cylinder block coolant temperature unchanged. Clough [8] conducted an experimental study on a 4 valves Jaguar gasoline engine. Some modifications were made on the block and head toward better cooling. Similar to Finlay et al. [4] he redesigned the cylinder head and block water jacket geometry to include higher coolant velocities in critical areas and adequate in other areas. He achieved higher BMEP (Brake Mean Effective Pressure), shorter warm up period and lower coolant pump power.

In the present work, both experimental and numerical techniques are used to investigate the water jacket of a four cylinders engine for achievement of almost the same flow rate at similar locations of the four engine cylinders. For example, the flow rate around the exhaust valves of the four cylinders should be similar. The Engine specification is given in Tab. 1.

Table 1 The Engine specifications

Specification	Value
Displacement volume (cm^3)	1649
Number of cylinder	4
Number of valves per cylinder	4
Bore (mm)	78,6
Stroke (mm)	85
Compression ratio	11,2

The original engine coolant passage along with its inlet and outlet had been designed unintentionally in such way that different coolant flow rate is obtained in similar regions of each cylinder. This fact leads to some problems when the engine fuel changes to natural gas or turbocharged. In these cases, due to the nonuniformity of the flow rate undercooling may take place at some critical areas of a cylinder while it might overcool the similar locations in the other cylinders of an engine. This study intends to make a minor modification in the flow pattern of the water jacket to reach a more uniform velocity through the water jacket passages. It should be noted that these modifications should be accompanied with minimum cost and geometry changes.

2 Numerical simulation

The coolant flow characteristics such as velocity, pressure drop and water jacket wall temperature must be identified carefully prior to doing any modification on the engine water jacket. Thus, the water jacket model of the original engine is first investigated numerically as one and three dimensional. The most critical thermal condition of the engine was considered in the thermo-fluid simulation. The engine at full load and 5500 rpm is chosen for this case as stated by [1, 9]. In this condition the volumetric flow rate of coolant is 120 l/min [9]. It should be noted that due to some limitations in the experimental set up the numerical simulation was validated for 65 l/min.

2.1 One dimensional simulation

GT Suit® software was used to simulate the water jacket flow one dimensionally as shown in Fig. 1.

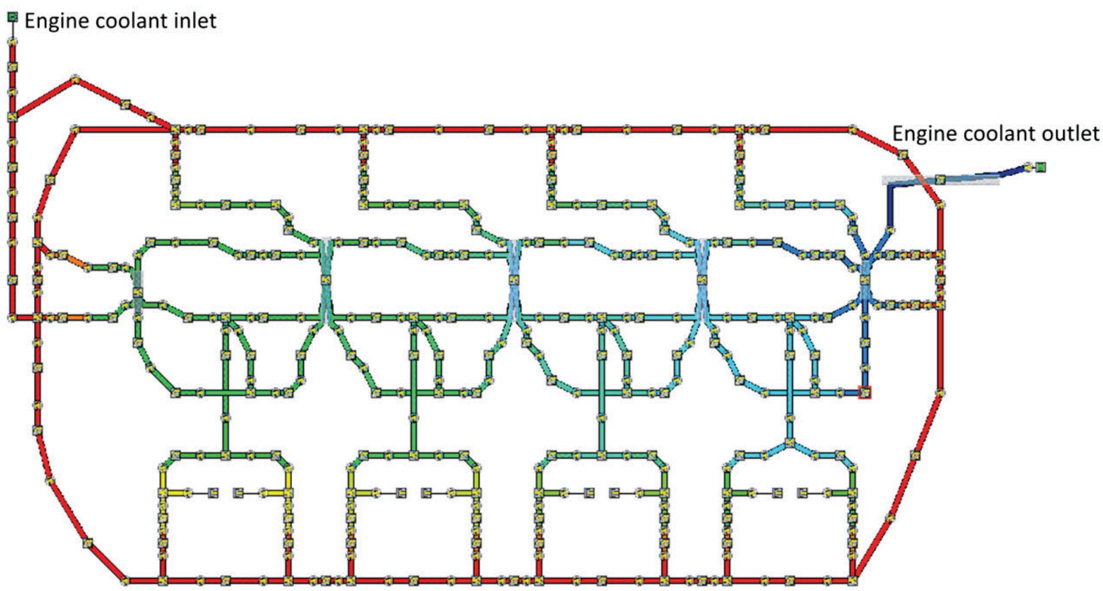

Figure 1 1D model of the engine water jacket passages

As shown in Fig. 1 all the cooling passages across the cylinder block and head were modelled as pipes, orifices and three ways. In this case the model was analysed only hydraulically to determine the velocity and pressure in each section. As an example the coolant velocity in the cylinder head passages is shown in Fig. 2.

2.2 Three dimensional simulation

The engine water jacket was modelled three dimensionally and meshed using Pro Engineering® and Hypermesh® software respectively. The steady thermo-fluid analysis of the whole engine (solid and fluid sections) was carried out numerically using the Fluent®

software. With the hydraulic analysis pressure velocity contour of the water jacket is obtained as shown in Fig. 3 and 4 respectively. The water jacket wall temperature is calculated based on thermal analysis as shown in Fig. 5.

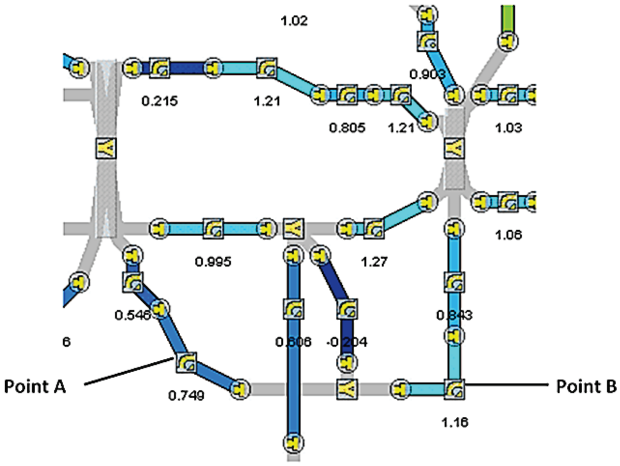

Figure 2 Coolant velocity calculated by 1D model of cylinder head No. 4 with 65 l/min

Figure 3 Pressure contour of the engine coolant at 65 l/min

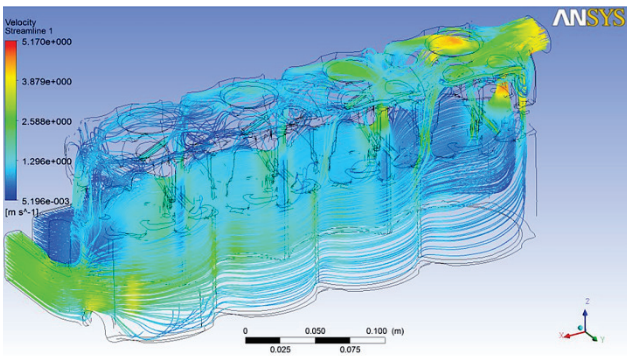

Figure 4 Velocity vectors of the engine coolant at 120 l/min

Figure 5 Temperature contour of the water jacket wall

3 Experimental simulation

The schematic of experimental set up is shown in Fig. 6. The experimental type of simulation was done for the numerical analysis validation as well as flow visualization within the cylinder head coolant passages for more detailed study.

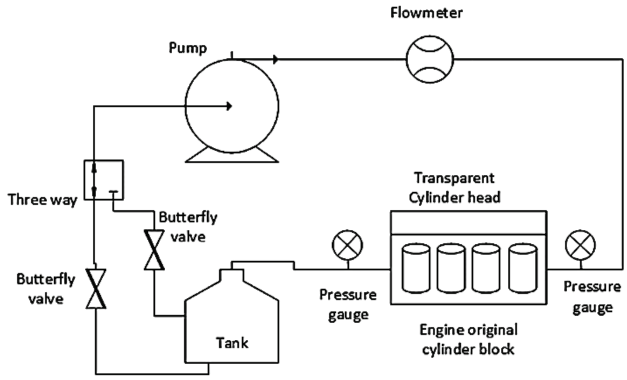

Figure 6 Schematic diagram of the experimental set up

The flow velocity and pressure drop through the water jacket are measured. A transparent cylinder head made of Plexiglas shown in Fig. 7 is used for local velocity measurement. This transparent cylinder head was installed on top of the original engine block as shown in Fig. 8.

Figure 7 A transparent cylinder head fabricated by plexy glass

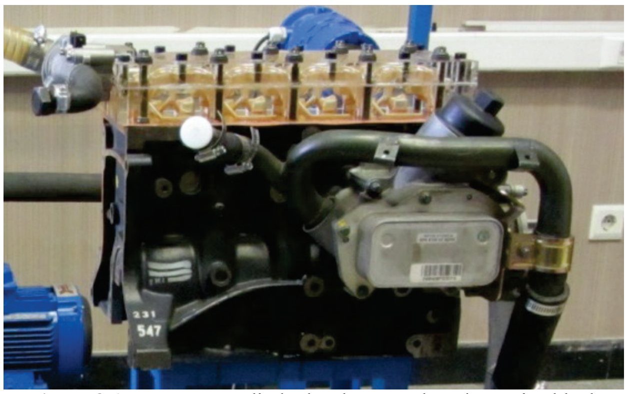

Figure 8 A transparent cylinder head mounted on the engine block

Particle Image Velocimetry (PIV) method is applied to measure the coolant velocity through the water jacket. PIV is an optical technique which is used for qualitative visualization and quantitative measurement of flow field such as velocity. This method gives an accurate and complete picture of the flow. For this purpose illuminated micron sized particle should be dispersed in the fluid and the travelled distance of them for a specific time period between two pulses of a light source is calculated. A laser beam is used for the light source. Some consecutive pictures are taken by a high speed camera. The pictures should be taken on the surface so a laser sheet is produced by passing the laser beam through the cylindrical prism as shown in Fig. 9 schematically [10].

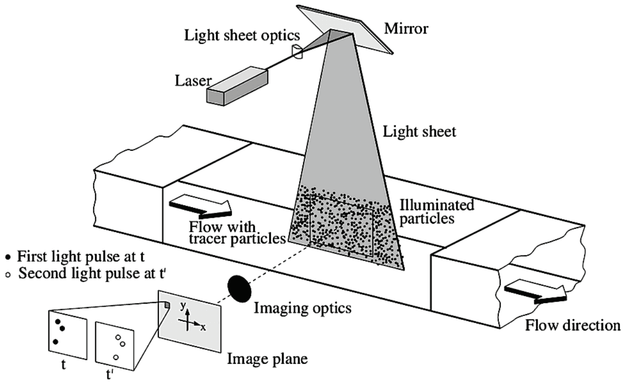

Figure 9 A schematic of PIV [10]

Particles with the diameter of 50 ÷ 75 μm are dispersed in the coolant. The specific gravity of particles applied for the experiment is about 1,15. A laser with the power of 5 Watt is employed to provide the laser beam. The motion blitz high speed camera is located above the cylinder head as shown in Fig. 10.

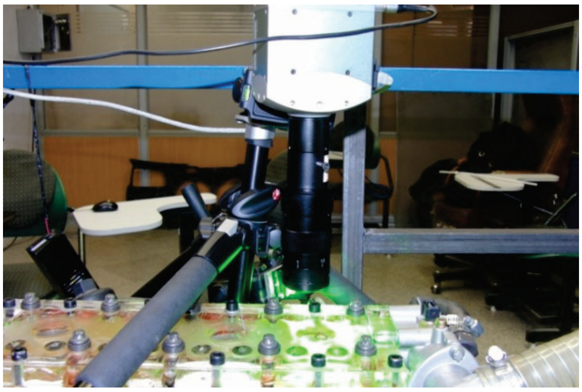

Figure 10 The location of high speed camera on the transparent cylinder head

Two local points of the cylinder head water jacket are selected for the velocity measurement as shown in Fig. 11 and 12. These points, shown by letters A and B, are chosen since the flow is almost two dimensional in these areas and also due to the PIV instrument limitation. Fig. 13 shows the location of these two points on a water jacket 3D model for better visualization.

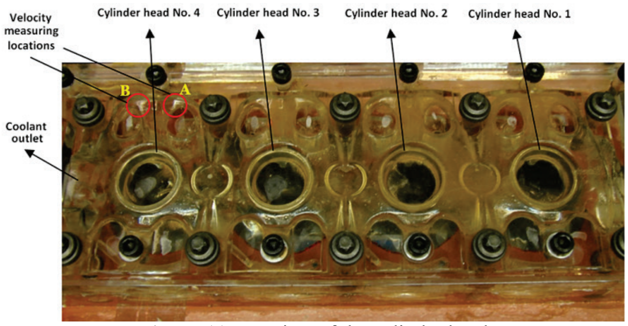

Figure 11 Top view of the cylinder head

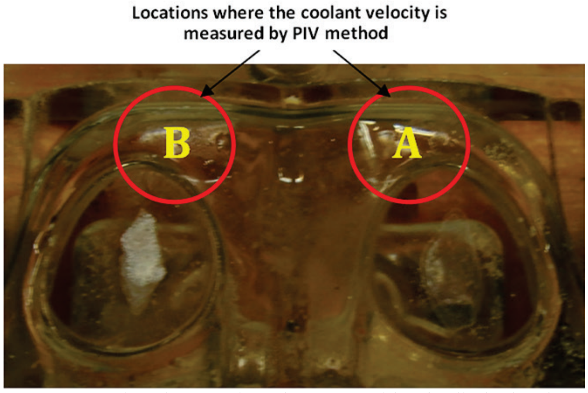

Figure 12 Two locations on the exhaust port side of cylinder head No. 4 where the coolant velocity is measured

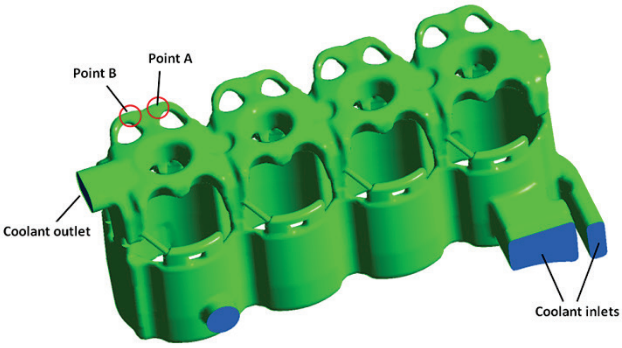

Figure 13 Two points where the velocity was measured on a 3D model

Picture frequency should be proportional to the flow velocity, laser source power, distance between the source and the location, etc. By taking the instrument locations, laser source power and the flow velocity into consideration the camera speed is set to 4000 frame per second for the best results. The pictures taken from each location A and B are shown in Fig. 14 and 15 respectively. These pictures are analysed by Matlab® developed image processing software, then their respective velocity contour of each location A and B is calculated and shown in Fig. 16 and 17. Pressure drop along the coolant passage is measured by installing two pressure transducers at the inlet and outlet of engine coolant passage.

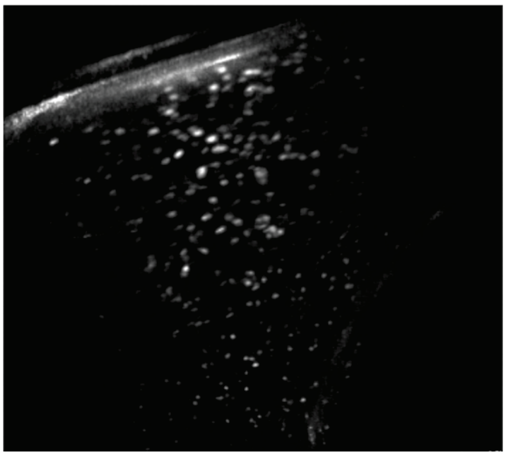

Figure 14 The picture taken from location A

The two parts determined by letters A and B in Fig. 2 are the equivalence one dimensional pipes of the same two regions A and B chosen in the cylinder head. The numbers next to each part represent its flow velocity. Similarly Fig. 18 shows the three dimensional model of these two locations along with their respective velocity vectors.

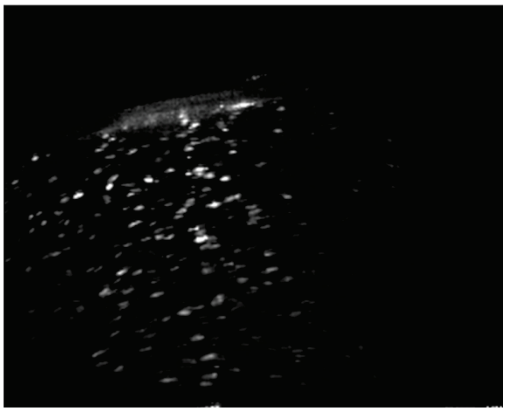

Figure 15 The picture taken from location B

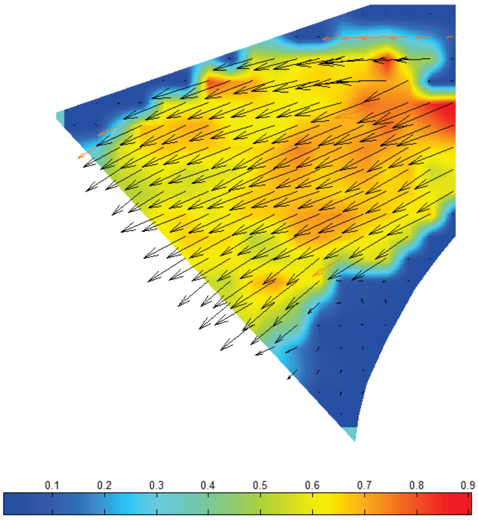

Figure 16 The image processing of the picture taken from location A

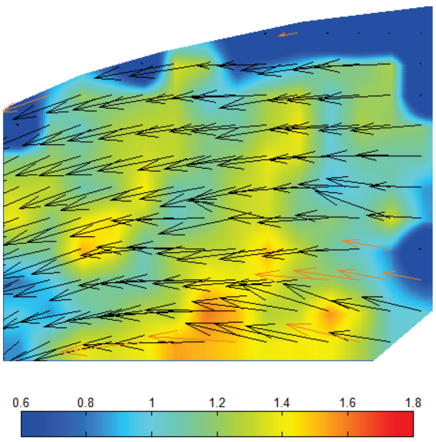

Figure 17 The image processing of the picture taken from location B

The calculated and measured velocity of each point A and B and the pressure drop of the water jacket are given in Tab. 2.

As given in Tab. 2 both 1D and 3D calculated values agree with the experimental values with good accuracy.

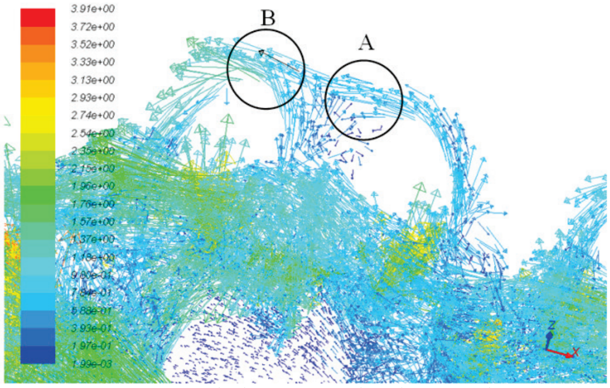

Figure 18 Velocity vectors at locations A and B

Table 2 Comparison of experimental measured data with the numerical solutions

	Experimental	1D model	3D model	1D Error	3D Error
Velocity of coolant at location A (m/s)	0,7	0,75	0,78	7	11
Velocity of coolant at location B (m/s)	1,2	1,16	1,1	3,3	8,3
Pressure drop along the water jacket (kPa)	10	8,7	8,6	13	14

Based on these numerical simulation validations and further analysis, it is observed that some parts of the engine water jacket are overcooled while there are some other parts threatened by high heat flux and insufficient cooling as shown in Fig. 19. As seen, the overcooling occurs at cylinder block No. 1 and 2 while at the same time some parts of their corresponding cylinder heads are experiencing very high temperature. This is due to the low coolant flow rate in these regions and can cause irreparable damage to cylinder head body. Therefore, some changes in the water jacket seem inevitable to create more uniform cooling along it.

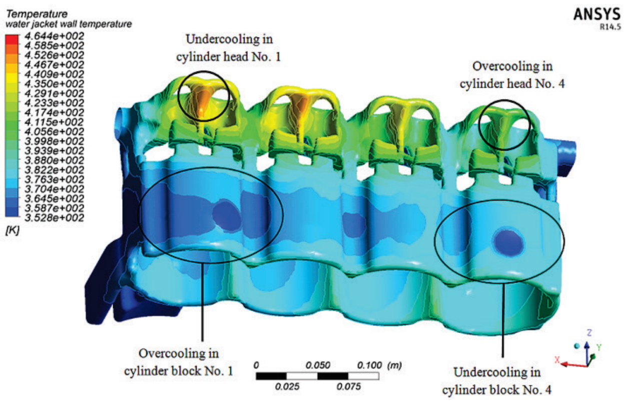

Figure 19 Nonuniform cooling along the water jacket

4 Case study for the engine cooling improvement

The primary purpose of this research is to minimize any existing improper cooling area as stated in the previous section with minimum engine coolant modification. This modification is focused only on the inlet and outlet patterns of the coolant for its simplicity, low cost and easiness. Three different strategies are proposed for creating more uniform cooling along the water jacket. Again, these strategies are just based on the different combination of inlet and outlet patterns and do not deal with the geometry of the coolant passages. In other words, the impact of different inlet and outlet combinations on the water jacket thermo-fluid is investigated.

Considering Eq. (1), the heat transfer coefficient depends on the thermo-physical properties of coolant as well as on the geometrical properties namely, hydraulic diameter and velocity [11].

$$Nu = \frac{h \cdot D}{k} = Nu(Re, Pr). \tag{1}$$

Since the thermo-physical properties of the coolant (because of low temperature gradient along the water jacket) and the internal geometry of the coolant passages of an engine remain constant then the heat transfer coefficient depends on the coolant velocity only. Thus the coolant velocity should be simply increased for the undercooled area and decreased for the overcooled region. This can be achieved mainly by hydrodynamic analysis.

Due to the good accuracy, simplicity and fast convergence of one dimensional model as explained above, it is used here to investigate the three proposed strategies for creating more uniform cooling along the water jacket.

Fig. 20 shows the original condition of coolant inlet and outlet and Figs. 21 to 23 show three different proposed configurations in comparison to its original one.

The flow rate of coolant for each case is tabulated in Tab. 3.

Table 3 The engine coolant inlet volumetric flow rates (l/min) for each strategy

	Original inlet	Head inlet	Cross inlet No. 1	Cross inlet No. 2	Cross inlet No. 3
Original	120	------	-------	-------	-------
1st Strategy	80	40	-------	-------	-------
2nd Strategy	80	20	9	7	4
3rd Strategy	80	20	9	7	4

Note that the 2nd and 3rd strategies are the same except the third one which has an extra outlet at the bottom of the main outlet in the block cylinder.

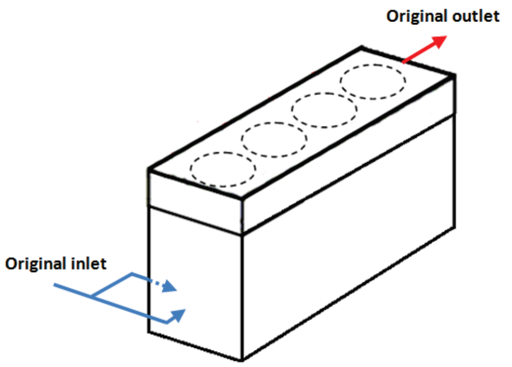

Figure 20 The original pattern of coolant inlet and outlet

From the thermo-fluid point of view cylinder number 1 and 4 block and head of the original engine are highly questionable as stated in the above analysis. The mass flow rate of the coolant around cylinder number one block is the most and the least around its head. The opposite of this takes place for cylinder number four. These high or low coolant flow rates yield over or under cooling. Therefore, the one dimensional analysis for the three different cooling strategies is carried out for the engine with keeping eyes on these two cylinders. Fig. 24 shows one dimensional coolant loop model of a cylinder block and head. The calculated coolant flow velocities for each three different cases are tabulated in Tab. 4 and 5 for cylinders 1 and 4 respectively.

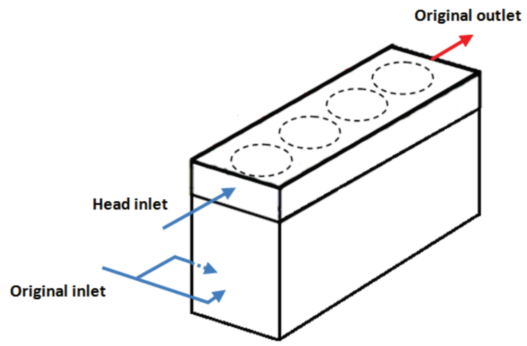

Figure 21 First strategy for coolant inlet and outlet pattern (an extra inlet for the whole cylinder head)

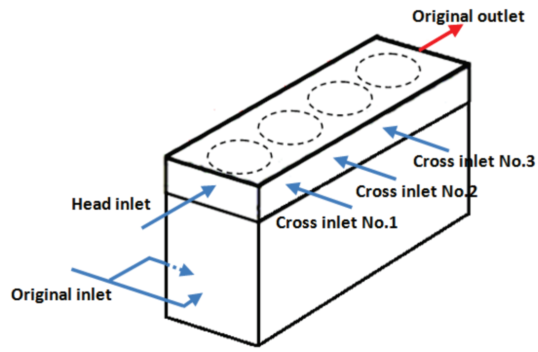

Figure 22 Second strategy for coolant inlet and outlet pattern (five extra inlets, one for the whole cylinder head and one for each cylinder head)

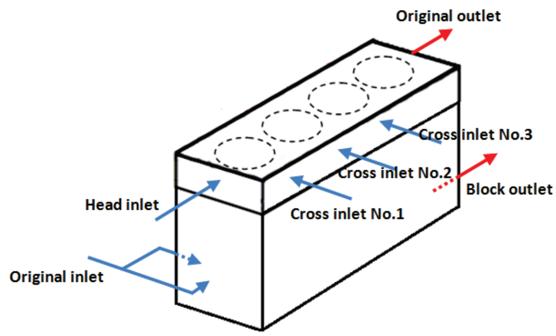

Figure 23 Third strategy for coolant inlet and outlet pattern (one extra outlet at last cylinder block and five extra inlets, one for the whole cylinder head and one for each cylinder head)

The number and letter notation used in these tables show the cylinder and location on it. For example, 4-Q means the point Q on the cylinder number 4. In Fig. 24 the letters P and Q are representatives for cylinder block passages. The letters A and B are representatives for coolant passages around the exhaust valves, letters C and D are representatives for lower and upper coolant passage between exhaust valves, letters E and F are representatives for coolant passages between exhaust and intake valves and letter H is representative for coolant passage around intake valves. Data related to cylinder numbers 2 and 3 are in between these ranges and because of its tediousness are not given here.

The strategies 1 and 3 show an appropriate coolant velocity reduction in the 1-P and 1-Q areas where overcooled originally. This reduction is not significant in the second strategy. As mentioned above the primary purpose of the proposed different cases were not only to decrease coolant velocity in the engine block 1 and 4 but also to provide an increment in their cylinder head coolant velocity. All three proposed cases increase the coolant velocity of cylinder head 1 at passages A and B but at the same time have opposite impact on the location C and D as shown in Tab. 4. The coolant flow through the passages below cylinder head was almost blocked and their velocities reach to around 0,02 to 0,03 m/s. This seems to occurs due to relatively high flow rate of the head inlet and also the intersection of these two flows. This low velocity may occur because film boiling might take place which must be avoided. On the other hand, in this strategy any coolant flow rate reduction through the cylinder head results in even lesser coolant velocity which can count as the main weak point of the first strategy.

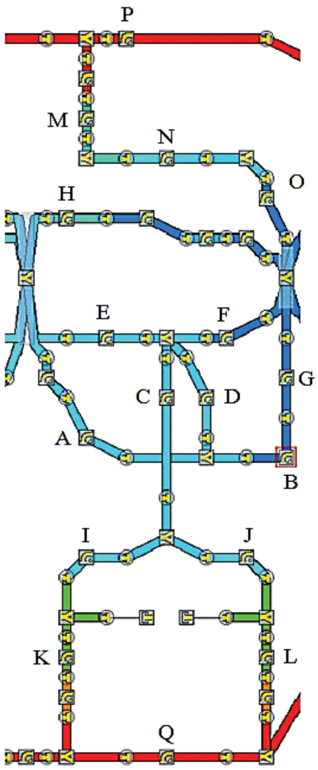

Figure 24 One dimensional cylinder coolant passage model along with the reporting velocity points

Table 4 Coolant velocity (m/s) for cylinder block and head No. 1

	1-A	1-B	1-C	1-D	1-E	1-F	1-G	1-H	1-I	1-J	1-K	1-L	1-M	1-N	1-O	1-P	1-Q
Original	0,94	1,38	0,93	0,22	0,87	1,33	1,01	0,23	2,6	2,6	1,34	1,34	0,76	1,67	1,48	0,77	0,97
1st strategy	1,53	1,85	0,52	0,16	1,61	1,85	1,35	0,3	1,47	1,47	0,75	0,75	0,45	0,99	0,87	0,6	0,74
2nd strategy	0,95	1,5	1,23	0,28	0,86	1,46	1,09	0,25	1,95	1,95	1	1	0,61	1,34	1,19	0,68	0,84
3rd strategy	1,16	1,48	1,16	0,3	0,86	1,43	1,07	0,24	1,18	1,18	0,61	0,61	0,4	0,91	0,79	0,6	0,74

Table 5 Coolant velocity (m/s) for cylinder block and head No. 4

	4-A	4-B	4-C	4-D	4-E	4-F	4-G	4-H	4-I	4-J	4-K	4-L	4-M	4-N	4-O	4-P	4-Q
Original	1,49	2,29	1,18	0,4	1,98	2,52	1,67	0,42	3,3	3,3	1,71	1,71	0,92	2,03	2,39	0,58	0,3
1st strategy	1,69	2,4	0,97	0,35	2,27	2,7	1,75	0,46	2,68	2,72	1,38	1,4	0,76	1,67	2,58	0,46	0,27
2nd strategy	1,57	2,34	1,11	0,38	2,1	2,61	1,71	0,44	2,94	2,98	1,5	1,53	0,83	1,84	2,48	0,5	0,29
3rd strategy	1,4	2,02	1,09	0,32	1,97	2,43	1,66	0,43	2,41	2,52	1,24	1,3	0,7	1,54	2,41	0,63	0,38

Comparison of the three strategies with the original one for cylinder number 4 implies that the 3rd strategy creates larger velocity in the 4Q and 4P areas than the original one as seen in Tab. 5. Although the 2nd and the 3rd strategy yield some reduction in the coolant velocity of the cylinder block number 4, obviously the 3rd one is more favourable. The main difference between the 3rd strategy and the 2nd one is an extra outlet which is located at the bottom of the original outlet in the block cylinder. This extra outlet seems causing the coolant flow velocity reduction in the cylinder head No. 4 and an augmentation in the block which both are constructive for this engine cooling. It appears that the strategy number three provides the most uniform velocity along the water jacket among the three proposed cases.

Table 6 The coolant pressure drop of the three proposed strategies along with its original one

	Original	1st strategy	2nd strategy	3rd strategy
Pressure drop along the water jacket (kPa)	36	27	31	24

In addition, the pressure drop of the coolant flow through the water jacket was calculated by 1D simulation and the results are tabulated in Tab. 6.

As given in Tab. 6, the minimum pressure drop belongs to the 3rd strategy. Eq. (2) which is used for water pump power indicates that lower pressure drop means lower input power to the pump.

$$\dot{W} = \frac{\dot{m} \cdot \Delta P}{\rho}. \qquad (2)$$

Thus, from both viewpoints, namely having more uniform velocity distribution and lower pressure drop, the strategy number three can be proposed as the new coolant inlet and outlet pattern for the engines.

5 Conclusion

The primary purpose of this work was to overcome any existing engine coolant flow pattern flaw with minimum modification. Thus, a case study was conducted to investigate the impact of inlet and outlet pattern on the coolant velocities at different regions along the water jacket of the internal combustion engine. This was done experimentally and numerically. A transparent cylinder head made of Plexiglas was mounted on the original cylinder block for the flow visualization and measurement. Particle Image Velocimetry (PIV) method was applied for these measurements. Along with the experimental work a comprehensive 1D and 3D numerical simulations were carried out to locate the inadequate cooling areas and then used for further case study to overcome those areas. Three new inlet and outlet coolant patterns were proposed and were simulated by numerical method after validation. The Numerical calculation showed that the third case that includes four extra inlets and one extra outlet creates more uniform velocity and also lesser presser drop across the water jacket in comparison with the original one. The coolant velocity reduction in overcooled regions and augmentation around the undercooled regions yield better engine heat transfer and performance.

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Nomenclature

D – Diameter, m
Pr - Nondimensional Prandtel number, -
h - Heat transfer coefficient, W/(m^2·K)
Re - Reynolds number, -
k - Thermal conductivity, W/(m·K)
$\dot{W}$ - Water pump power, kW
$\dot{m}$ - Mass flow rate, kg/s
P - Pressure, Pa
Nu - Nondimensional Nusselt number, -
ρ - Density, kg/m^3

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