



Integration of exhaust manifold with engine cylinder head towards size and weight reduction

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ABSTRACT

In this research, a new exhaust manifold and its cooling jackets is first designed for the integrated exhaust manifold into cylinder head (IEMCH) for a turbocharged engine. Then, the gas exchange and flow analysis is carried out numerically to evaluate the proper conditions for the exhaust gas and the coolant stream respectively. Finally, the entire engine parts are thermally analyzed to assure their acceptable temperature. The obtained results are compared to the base engine conditions, indicating that the new engine with IEMCH may well satisfy all its thermal limitations.

1) Introduction

Substantial merits will be achieved by integrated exhaust manifold into cylinder head (IEMCH) versus the conventional internal combustion engines. In conventional engines, exhaust manifold is an independent component, made of cast iron. To enhance its thermal resistance in a turbo-charged engine, expensive alloys are combined with the cast iron. On the other hand, for the IEMCH, the exhaust manifold is manufactured as one part with the cylinder head. Thus its material changes from cast iron to aluminum which has much lower thermal resistance than cast iron. The IEMCH not only lower the manufacture cost but also it has many advantages such as, lower weight and volume, less fuel consumption and faster warm up duration. But the exhaust manifold must be cooled due to its lower thermal resistance.

The exhaust manifold of conventional turbocharged engines is an individual component which is made of the cast iron composed of nickel alloy and is mounted to the cylinder head as shown in Figure 1(a). The nickel alloy is combined with the material of exhaust manifold in turbocharged engines to improve its thermal resistance against the adjacency to the hot exhaust gas. The maximum temperature of exhaust gas is typically 1050°C [1-3] which is much higher than that of the natural aspirated engines. There is a stainless steel gasket between the exhaust manifold and the cylinder head to avoid the gas leakage. In the IEMCH, the exhaust manifold is a part of the cylinder head as shown in Figure 1(b) [3]. Therefore, the material of the cylinder head and the new exhaust manifold in the IEMCH must be the same, such as aluminum. Hence, the newly designed cylinder head does not require the exhaust manifold gasket [3]. Some heat flux is removed from the new cylinder head by means of exhaust manifold cooling, to assure that the new used material for the exhaust manifold remains in the range of acceptable temperature.

The lower density of replaced aluminum ($\rho=2700 \text{ Kg/m}^3$) versus that of the cast iron ($\rho=7800 \text{ Kg/m}^3$) [4-5] makes the new cylinder head about 3.2 Kg lighter and 11 millimeters smaller in the IEMCH as shown in Figure 2 [3]. The path of the exhaust gas to the turbine decreases according to Figure 3 [3].

The lighter and downsized cylinder head was also indicated as an important benefit by Borrman *et al.* [1-2] and Coltman *et al.* [6]. They integrated the exhaust manifold into the cylinder head for a turbocharged engine. There is no exhaust gas leakage between the cylinder head and the exhaust manifold in the IEMCH and the exhaust manifold gasket is eliminated completely [3].

Due to lower heat capacity ($C=2430 \text{ kJ/K}$) and higher conductivity ($k=237 \text{ W/mK}$) of aluminum (results higher thermal diffusivity) versus those of the cast iron ($C=3354 \text{ kJ/K}$, $k=60 \text{ W/mK}$) [4-5], the IEMCH has a faster warm-up period at the cold start condition [3]. This leads to shorter light-off time of the catalyst convertor [7]. The IEMCH either in a turbocharged engine, as described in this paper and also by Borrman *et al.* [1-2] and Coltman *et al.* [6] or in a natural aspirated one by Kojima [8] leads to the faster warm-up time and lower catalyst converter light-off time at the cold start duration [3].

The lower manufacture cost which is one of the most important benefits of the IEMCH is achieved by elimination of extra parts, such as the exhaust manifold gasket, 7 nuts and 7 bolts [3]. Also, there is no need to nickel material, and the time for assembling process reduces significantly [1-3,6-7].

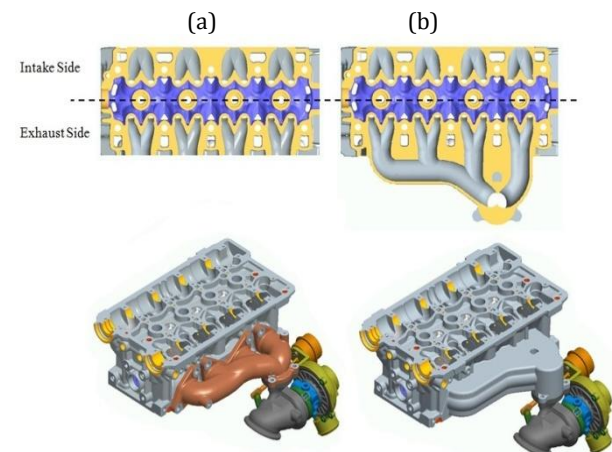


Figure 1: Cylinder head and exhaust manifold of turbocharged EF7 engine, (a) base engine, (b) IEMCH

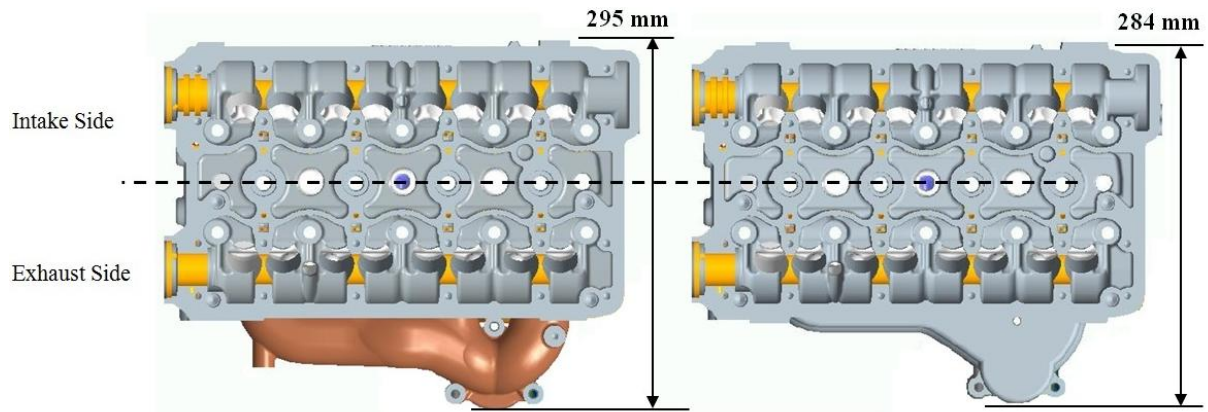


Figure 2: The total widths of cylinder head and exhaust manifold, (a) Base engine, (b) IEMCH, the IEMCH has been downsized

Main portion of the exhaust manifold manufacture cost is allocated to the nickel supply, due to its price that is about 40 \$/Kg in 2010, and approximately 3 to 4 kilograms nickel is needed for a turbocharged engine exhaust system [1,3]. Removing the over-fuelling at full load and high speed conditions is another consequence of the IEMCH, over-fueling process is performed in the base engine to control the exhaust manifold temperature by evaporative cooling [1-3,6,8]. The cooled exhaust manifold removes the over-fuelling and its mechanisms, either in integrated cylinder head as proved by Borrman *et al.* [1-2], Coltman *et al.* [6] and Kojima [8], or in the independent one by Ito *et al.* [9] and Taylor *et al.* [10].

Borrman *et al.* [1-2] showed experimentally that the exhaust gas temperature on the IEMCH is lower than the base engine. Figure 3 indicates that the rate of the new exhaust manifold cooling increases from 45% to 70%. This causes the peak temperature of the new one decrease at full load conditions significantly. Hence, it improves the catalyst converter life time and its performance [3,6,10-12]. Economic fuel consumption and emissions reduction are also some of the other advantages of this cylinder head [3]. Lower frictional losses due to the faster warm-up time at cold start condition and the elimination of the over-fuelling at full load condition, results lower fuel consumption [1-3,6]. Improving catalyst performance, the faster warm-up time and preventing gas leakage from the combustion chamber and the exhaust manifold gasket contribute to the lower pollutant emissions [1-3,6,13]. The 2 to 3 millimeters clearance between the cylinder head and exhaust manifold in the base engine is a main source of emission. In addition, the fewer number of parts and the water stream around the exhaust runners reduce the noise level, vibration and harshness (NVH) [3].

The only disadvantage of IEMCH is controlling the temperature of the new exhaust manifold properly which has lower thermal resistance versus that of the cast iron [3]. As a result, the cooling water jackets are designed around the exhaust runners

according to Figure 3 and the new cooling system includes some changes consequently [3]. The later changes have also indicated by Borrman *et al.* [1, 2], Coltman *et al.* [6], Ito *et al.* [9] and Taylor *et al.* [10]. In this research, the IEMCH is applied to the turbocharged EF7 engine. The specifications of this engine are given in Table 1 [14]. New exhaust runners along with the new cooling jackets are designed. Numerical analysis is carried out in the exhaust manifold and the water jacket to investigate its modifications. Then, thermal analysis is performed to assure adequate thermo-fluid conditions at the water jacket and the engine structure eventually. All the numerical thermo-fluid analysis is first performed on the base engine for comparison and validation of the obtained results [5].

2) Gas exchange analysis

The new 3 dimensional IEMCH model is created by ProEngineer software [5]. The new exhaust runners have shorter paths to the turbine and smaller cross sections in the IEMCH compared to those of the base engine [3]. Borrman *et al.* [1-2] indicated that the turbine performance and the engine layout are improved by lowering the length of exhaust runners. Figures 3 and 4 show that there is some reduction on the new exhaust runners length and cross sections respectively.

Since there is no experimental data for the integrated engine, the numerical analysis was carried out for the base engine first for comparison with the existing experimental data [5]. The experimental data for the base engine (Turbocharged EF7) were collected in Irankhodro Powertrain Company (IPCO). These data are tabulated in Table 2 [14] and compared with the present numerical results. As indicated in Table 2 the maximum error between these two is about 7% and their averaged error is about 2%.

The equivalent heat transfer circuit for the exhaust manifold is shown in Figure 5. Thermal resistance of the exhaust manifold is calculated by Equation 1 [3-5] and the values listed in Table 3.

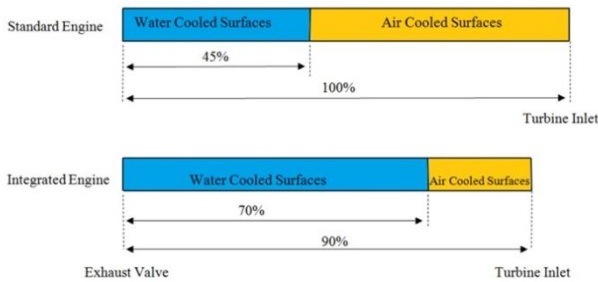


Figure 3: The schematic of the exhaust runners and cooling surfaces before turbine

Table 1: The specification of EF7 Engine [14]

No. of cylinders	4
Bore	78.6 (mm)
Stroke	85 (mm)
Compression ratio	9.8
Displacement	1648 (cm ³)
No. of valves	16 (4 per cylinder)
Max. power	110 (kW) at 5500 (rpm)
Max. torque	215 (Nm) from 2200 to 4800 (rpm)
Environmental standard	Euro 4
Firing order	1-3-4-2
Fuel	Gasoline (based on natural gas)

The calculated results point out that there is a significant reduction, about 13 times, on the thermal resistance of the new exhaust manifold with respect to the base one.

Therefore, more heat flux is rejected from the heat transfer surfaces.

$$\sum R = R_1 + R_2 + R_3 = \frac{1}{A} \left[\frac{1}{h_g} + \frac{(r \ln \frac{r_o}{r_i})}{K} + \frac{1}{h_\infty} \right] \quad (1)$$

To prove that there is adequate thermo-fluid conditions for the exhaust gas in the exhaust runners of the IEMCH, gas exchange analysis is carried out numerically in both engines. HYPERMESH is the selected software to create appropriate mesh for the exhaust manifold. The mesh size of the exhaust runners is approximately 2 millimeter producing 219771 tetrahedral cells throughout the geometry as shown in Figure 6, with the minimum and maximum angle about 23 and 127 degree respectively. The analysis is checked for mesh size independency and 2893 elements selected. The boundary conditions of the numerical study consist of variable mass flow rate of exhaust gas with variable temperature in accordance with crank angle of the engine. These data is extracted from GT-POWER one dimensional software. The upstream turbine section is considered pressure outlet and also the inner surface of exhaust manifold is referred as wall [5] illustrated in Figure 6. Figure 7 shows the results of the study, velocity distribution contours as a function of degree crank angle for the both base engine and the IEMCH.

Table 2: Comparison between experimental data and the results of numerical analysis for the base engine [14]

Data	Calculated	Experimental	Error (%)
TDW ¹	5.37 (°C)	5.16 (°C)	4.07
ETUT ²	927.66 (°C)	927.35 (°C)	0.03
EGT ³ on Runner No.1	869.88 (°C)	889.92 (°C)	2.25
EGT on Runner No.2	896.09 (°C)	927.48 (°C)	3.38
EGT on Runner No.3	918.67 (°C)	913.89 (°C)	0.52
EGT on Runner No.4	898.79 (°C)	909.90 (°C)	1.22
EPUT ⁴	244.75 (KPa)	263.60 (KPa)	7.15
CP ⁵ after Water Pump	88.69 (KPa)	89.54 (KPa)	0.95
CP before Thermostat	59.56 (KPa)	61.93 (KPa)	3.82
TCHM1-1 ⁶	223.16 (°C)	231.90 (°C)	3.77
TCHM1-2 ⁷	221.59 (°C)	228.31 (°C)	2.94
TCHM1-3 ⁸	249.69 (°C)	238.17 (°C)	4.84

¹Temperature difference of water jacket (between the inlet and outlet)

²Exhaust temperature upstream turbine

³Exhaust gas temperature

⁴Exhaust pressure upstream turbine

⁵Coolant pressure

⁶Temperature of cylinder head material on cylinder 1, between exhaust valves and spark plug

⁷Temperature of cylinder head material on cylinder 1, between exhaust valves

⁸Temperature of cylinder head material on cylinder 1, between exhaust valves

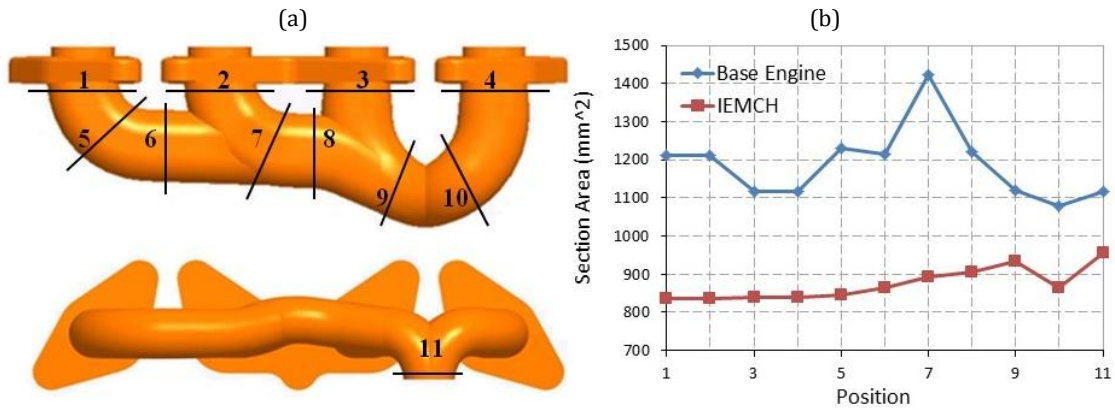


Figure 4: Exhaust runners (a) geometry, (b) cross section areas

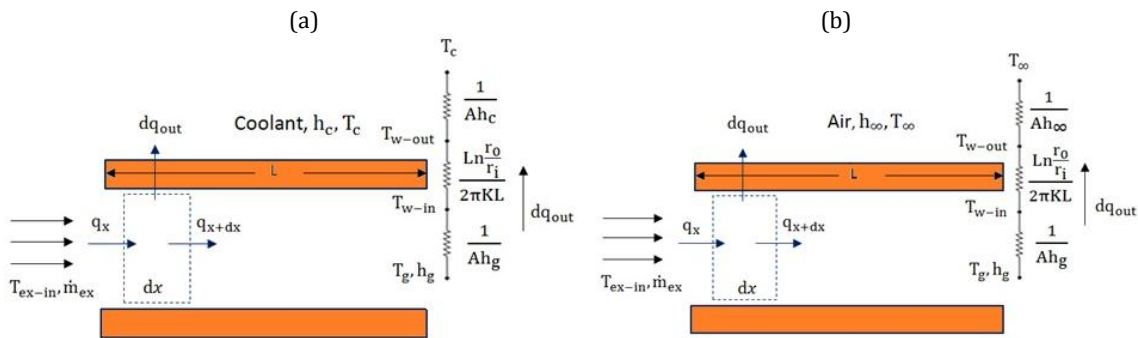


Figure 5: Equivalent heat transfer circuit for the exhaust manifold, (a) Base engine, (b) IEMCH [3]

Table 3: Parameters values used in equation 1 [3-5]

Parameters	Base engine	IEMCH
T_{∞}	30 (°C)	88 (°C)
A	49800 (mm ²)	39760 (mm ²)
h_g	400 ($\frac{W}{m^2 \cdot K}$)	472 ($\frac{W}{m^2 \cdot K}$)
r_i	190 (mm)	160 (mm)
r_o	195 (mm)	165 (mm)
K	60 ($\frac{W}{m^2 \cdot K}$)	237 ($\frac{W}{m^2 \cdot K}$)
h_{∞}	30 ($\frac{W}{m^2 \cdot K}$)	7500 ($\frac{W}{m^2 \cdot K}$)
ΣR	0.3	0.023

The exhaust gas velocity and temperature at the upstream turbine as a function of crank angles for the base engine and the integrated one are shown in Figures 8 and 9 respectively. The new exhaust gas experiences higher velocity due to its smaller cross section and lower temperature (about 50°C to 60°C) by means of cooling as Borrmann et al. [1-2] shown experimentally. Thus, heat rejected of the exhaust gas is improved from 0.9 kilowatt to 10.5 Kw [3,5]. On the other hand, the waste exhaust heat calculated by Equation 2 [4] is lessened from 145 kilowatt to 135.8 Kw in the IEMCH [5]. The exhaust manifold temperature and its thermal durability are controlled by the cooling properly. The inner and outer wall temperatures of the exhaust manifold at

a section 10 centimeters away from the third runner entrance, the section 3 in Figure 4-a, are shown in Figures 10 and 11 respectively. The inner wall temperature is lowered from 790°C in the base engine to 160°C in the IEMCH. Furthermore, it could be noticed that the exhaust manifold temperature fluctuations are almost eliminated which reduces thermal stresses significantly. Borrmann et al. [1-2], Coltmant et al. [6], and Ito et al. [9] also indicated lower thermal stresses and higher reliability of IEMCH parts.

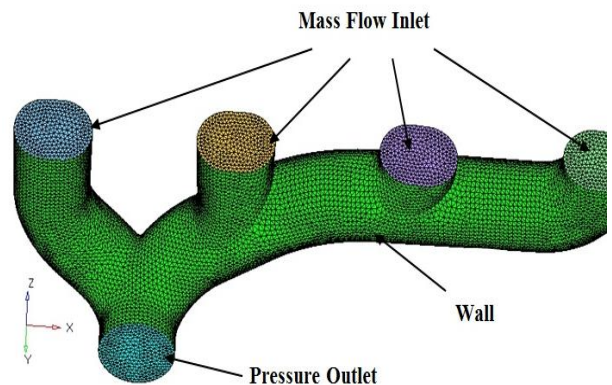
$$Q_{ex} = \dot{m}_{ex} C T_{ex} \tag{2}$$


Figure 6: Tetrahedral mesh element created through the geometry of exhaust runners by HYPERMESH software, exerted boundary conditions on the exhaust manifold

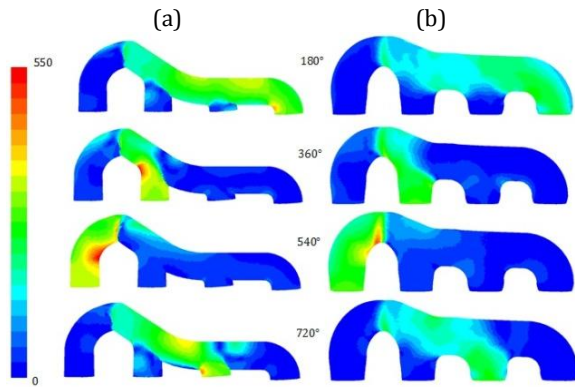


Figure 7: Exhaust gas velocity (m/s) for 180, 360, 540 and 720 degree crank angle; (a) base engine and (b) IEMCH

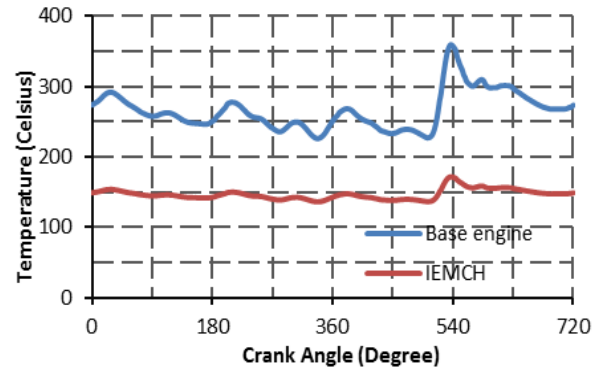


Figure 11: Outer wall temperature of the exhaust manifold at full load condition

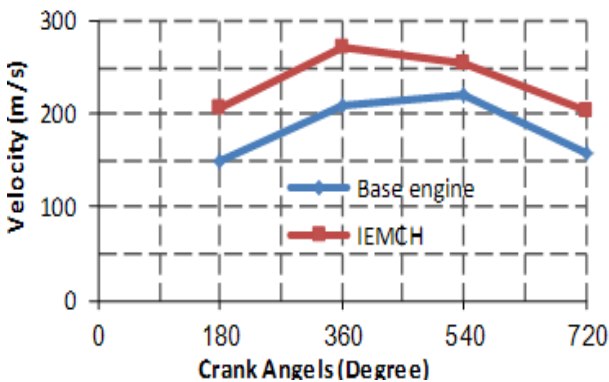


Figure 8: Exhaust gas mean velocity at upstream turbine at full load condition

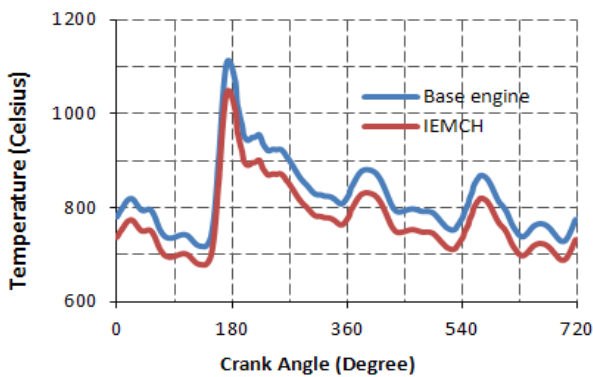


Figure 9: Exhaust gas temperature at upstream turbine in runner 1 at full load condition

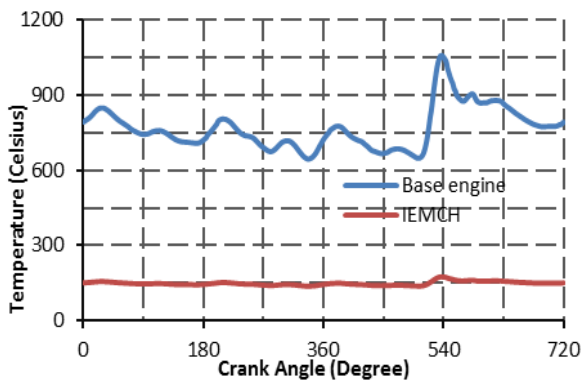


Figure 10: Inner wall temperature of the exhaust manifold at full load condition

3) Flow analysis

It is essential to cool the exhaust runners made of aluminum in the IEMCH, in order to provide an acceptable strength and durability for the materials against high temperature gases and thermal stresses in the exhaust manifold. The water jacket of cylinder head extends to the area of exhaust runners according to Figure 12 to maintain the materials in an acceptable temperature level. The cross sections of the coolant through the cylinder head gasket at the exhaust side are widened in order to provide sufficient cooling for the exhaust manifold [5]. Due to the pressure drop reduction between the cylinder head and the block, the mass flow rate through the water jacket should be increased. Hence, as Borrmann *et al.* [1-2] stated the water pump flow rate should be promoted to counteract it.

The flow analysis is carried out to assure sufficient cooling of the entire new cylinder head water jacket. The fluid flow analysis is numerically fulfilled by Computational Fluid Dynamics, Fluent software. In this process, the continuity and momentum equations [4] are simultaneously solved for the coolant as an incompressible fluid consists of 50% water and 50% ethylene glycol. The boundary conditions are related to the full load regimes which the water pump supplies 140 lit/min mass flow rate. Obviously, the boundary conditions consist of mass flow inlet and pressure outlet [5]. In order to validate the accuracy of the results, the flow analysis is first performed for the conventional water jacket engine. The velocity distribution of coolant in the cylinder head water jacket is illustrated in Figure 13, both for the base engine and the integrated one.

Since high temperature occurs near the exhaust valves and spark plug, special attention must be given on the cooling around these regions. The mass flow rate and mean heat transfer coefficient around the valves bridge of each cylinder are shown in Figures 14 and 15, respectively. As expected widening the cylinder head gasket passages leads to more mass flow rate in the exhaust side of the

cylinder head as shown in Figure 14. Consequently, the heat transfer coefficient increases near the exhaust valves. Figure 16 shows an improvement of the mean heat transfer coefficient in terms of water pump mass flow rate. The larger water jacket and the lower pressure loss between the cylinder head and the block is one of the main reasons for this phenomenon. The mean heat transfer coefficient in terms of the water pump mass flow rate is shown in Figure 17 which indicates an acceptable cooling around the exhaust runners.

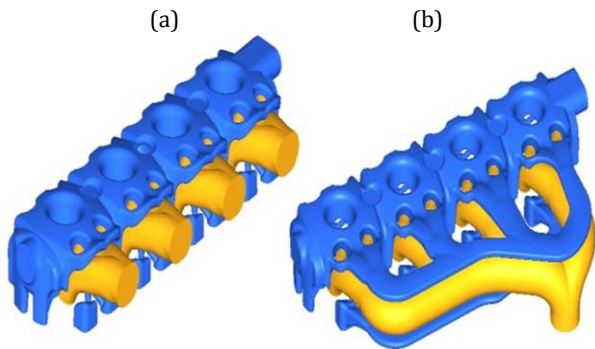


Figure 12: EF7 cylinder head water jacket with its exhaust ports; (a) base engine, (b) IEMCH

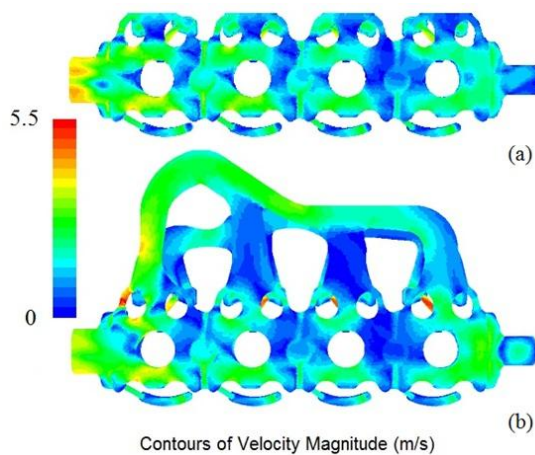


Figure 13: Coolant velocity distribution (m/s) in the cylinder head when the thermostat is fully opened; (a) base engine, (b) IEMCH

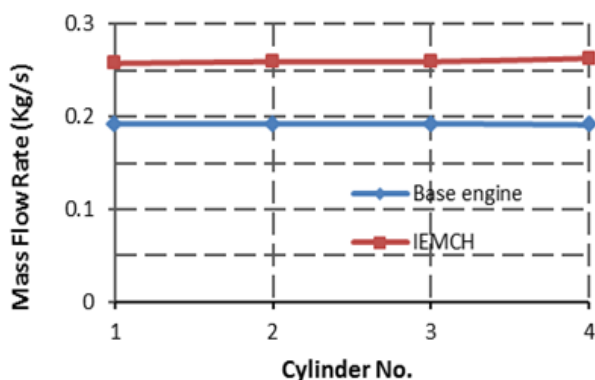


Figure 14: Coolant mass flow rate around the exhaust valves at fully thermostat opening

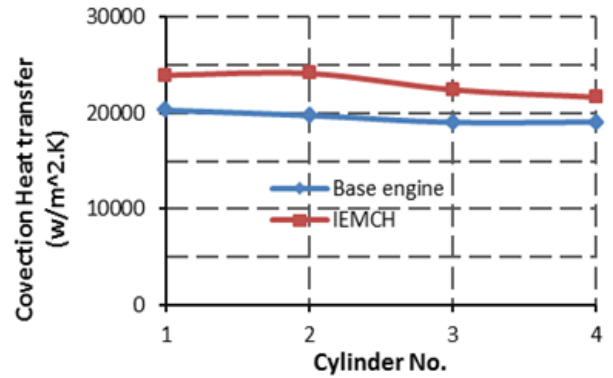


Figure 15: Mean heat transfer coefficient around the exhaust valves at fully thermostat opening

4) Thermal analysis

Thermal conditions of the IEMCH should be examined for its temperature limitations. The steady thermal analysis of the engine is numerically carried out by Fluent software for the solid part and the water jacket simultaneously at full load and high speed (5500 revolutions per minutes) conditions [5]. This is done to confirm the allowable temperature of the engine structure and coolant.

The heat flux to the cylinder walls is determined by convective heat transfer coefficient from Woschni [5,11,15-16], Equations 3 and 4, and some available experimental data from the manufacturer [14].

The inner surface heat flux of the exhaust manifold is determined by the heat transfer coefficients and the mean exhaust gas temperature which were obtained from the three-dimensional gas exchange analysis. The maximum temperature of the cylinder head structure occurs around the valves bridge and spark plug similar to the base engine.

In general, there is no significant temperature difference on the cylinder head between the base engine and IEMCH according to Figure 18. The obtained results indicates that the walls temperature of the cooled aluminum exhaust runners remain lower than the allowable value as shown in Figure 18. In the water jacket, the heat transferred to the coolant is increased from 55.3 kilowatt to 65.8 kW in the integrated engine which results 1°C raised on the outlet temperature. Hence, the water pump must be promoted to supply more mass flow rate to overcome this raise. Also, a larger radiator or fan is required to reject the extra heat from the coolant.

$$\bar{h}_{c,g} = \frac{1}{4\pi} \int_0^{4\pi} h_{c,g}(\theta) d\theta \quad (3)$$

$$\bar{T}_g = \frac{1}{4\pi h_{c,g}} \int_0^{4\pi} h_{c,g}(\theta) T(\theta) d\theta \quad (4)$$

Equation 5 stated by [12] is used here to calculate the energy balance of both engines.

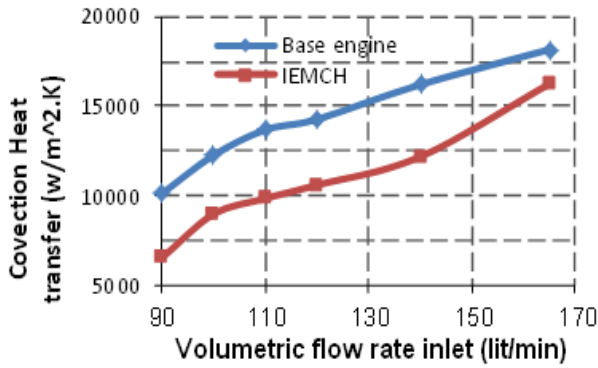


Figure 16: Mean heat transfer coefficient in cylinder head water jacket - at fully thermostat opening

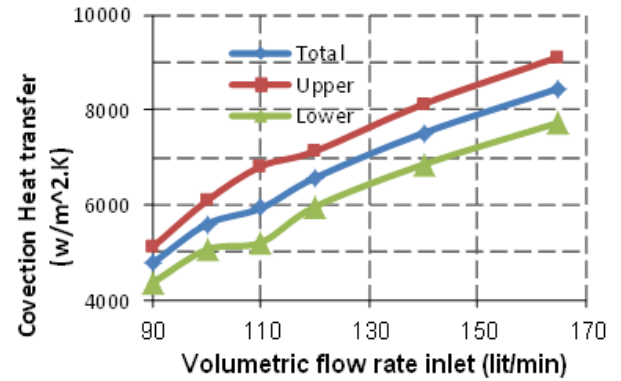


Figure 17: Mean heat transfer coefficient around exhaust runners for IEMCH- at fully thermostat opening

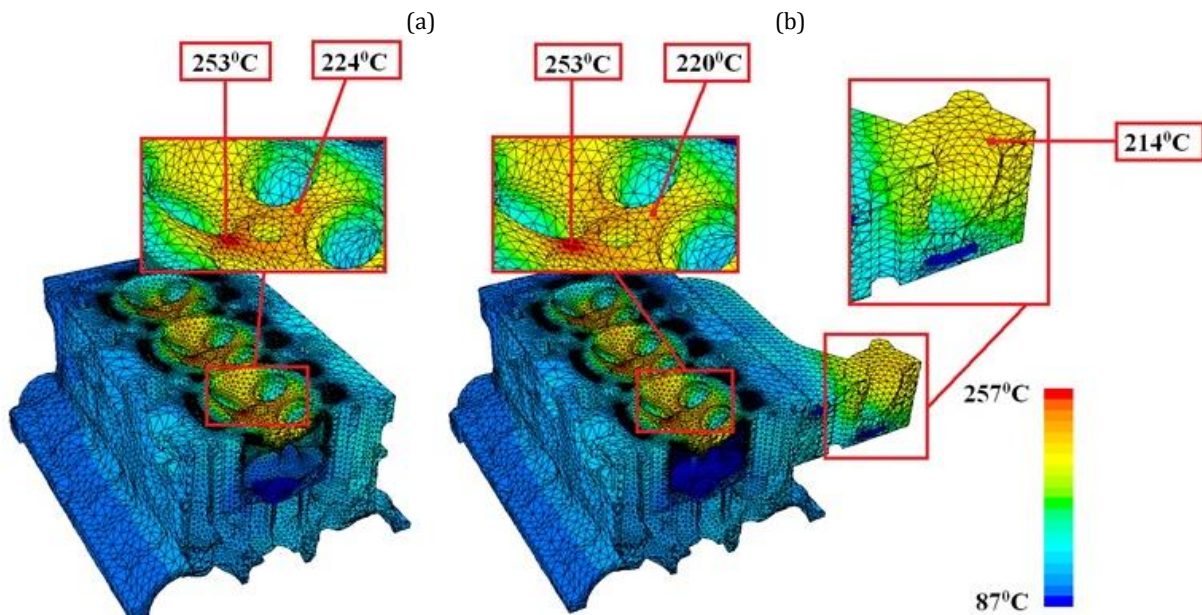


Figure 18: Cylinder head temperature distribution at full load and 5500 revolutions per minute, (a) base engine, (b) IEMCH

Figure 19 illustrates the energy balance in the integrated and base engines graphically. It is based on 100 units of the crankshaft power. It seems that the integrated engine is more isolated from its surrounding because the heat transfer to the ambient is decreased from 22 to 14, about 35% reduction. In addition, in the integrated engine the exhaust gas enthalpy is decreased from 132 kilowatt to 124 Kw which this reduction is somehow added to the coolant [5].

$$\dot{Q}_{amb} + \dot{Q}_{water} + \dot{Q}_{oil} + (\dot{m}h)_{ex} + \dot{Q}_{ex-Ch} + \dot{Q}_{ex-amb} + \dot{W}_{shaft} = \dot{Q}_{in} \quad (5)$$

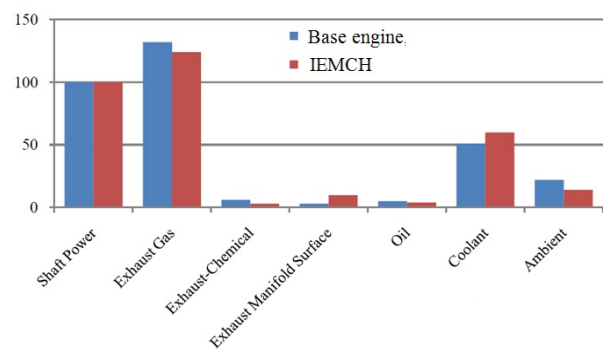


Figure 19: Engine energy balance at full load and 5500 revolutions per minute (a) base engine, (b) IEMCH

5) Conclusions

Thermo-fluid analysis is carried out on an Integrated Exhaust Manifold into Cylinder Head (IEMCH), in a turbocharged engine numerically to assure its proper performance and thermal limitations. Based on this study the following conclusions can be drawn:

- There is a weight reduction of 3.2 kilograms and better layout condition for IEMCH.
- Lower manufacture cost because of the elimination of expensive alloy, exhaust manifold gasket, bolts, and nuts.
- Economical fuel consumption.
- Emissions reduction due to the exhaust gas leakage prevention and improvement of catalyst performance.
- Higher heat transfer coefficient around the exhaust valves.
- Lower coolant heat transfer coefficient in the cylinder head.
- An improved water pump, a larger radiator and fan are required.
- The Maximum temperature of the exhaust gas at full load decreases about 55°C.
- The inner wall temperature of the exhaust manifold reduces from 790°C to 160°C.
- More heat flux to the coolant.

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Appendix

Notation

A	Area
BEV	Between exhaust valves
C	Specific heat capacity
Conventional	Base engine
CP	Coolant pressure
EGT	Exhaust gas temperature
EPUT	Exhaust pressure upstream turbine
ETUT	Exhaust temperature upstream turbine
h	Convection heat transfer coefficient
\bar{h}	Mean heat transfer coefficient
IEMCH	Integrated exhaust manifold into cylinder head
Integrated	Integrated exhaust manifold
K	Conductivity
L	Length
\dot{m}	Mass flow rate
q	Heat transfer rate
\dot{Q}	Heat transfer rate
r	Radius
R	Thermal resistance
T	Temperature

TCHM1- i	Temperature of cylinder head material on cylinder i , between exhaust valves
TDW	Temperature difference of water jacket (between the inlet and outlet)
\bar{T}	Mean temperature
x	Length
ρ	Density
θ	Crank angle

Subscripts

amb	Ambient
c	Coolant
c,g	Combustion chamber exhaust gas
$ex-amb$	Exhaust enthalpy to ambient
$ex-ch$	Exhaust chemical enthalpy
$ex-in$	Exhaust gas inlet
g	Gas
i	Inner
o	Outer
w	Wall
w,c	Wall adjacent to coolant
w,g	Wall adjacent to gas
w,in	Inner wall
w,out	Outer wall
∞	Air

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یکپارچه سازی چندراهه خروجی با بستار موتور برای کاهش اندازه و وزن

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کلیدواژه‌ها:

چندراهه خروجی یکپارچه با بستار

چندراهه خروجی خنک کاری شده

چندراهه خروجی آلومینیومی

تحلیل حرارتی سیالاتی

در این مطالعه، در ابتدا چندراهه خروجی جدید و راهگاه‌های خنک کاری آن برای چندراهه خروجی یکپارچه با بستار در موتور ملی پرخوران طراحی شد. سپس، تحلیل عددی تبادل گاز و خنک کاری انجام شد تا به ترتیب از شرایط مطلوب برای گازهای خروجی و جریان سیال خنک کننده اطمینان حاصل گردد. در نهایت، کل قطعات موتور تحلیل حرارتی شد تا شرایط دمایی قابل قبول آنها بررسی گردد. مقایسه نتایج در هر مرحله با موتور پایه نشان داد موتور جدید با چندراهه خروجی یکپارچه با بستار از محدودیت‌های حرارتی تجاوز نمی کند.

تمامی حقوق برای انجمن علمی موتور ایران محفوظ است.