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DESIGN OF AN INTERCOOLER OF A TURBOCHARGER UNIT TO ENHANCE THE VOLUMETRIC EFFICIENCY OF DIESEL ENGINE

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ABSTRACT

The commercially available automotive intercooler used to cool down the compressed hot air from turbo compressor fails when the summers are very hot and the speed of the vehicle is lowered. This is because, when speed of the vehicle gets reduced, the mass flow rate of atmospheric air in to the intercooler also get reduced and when the summers are hot, the temperature of atmospheric air becomes very high due to which the efficiency of intercooler get reduced. As a result of this, the density of air and consequently the mass flow rate delivered by the turbo compressor and entering into the engine get reduced. An intercooler is being designed here with proper heat transfer area to increase this mass flow rate by mixing some additional amount of conditioned air into the atmospheric air entering into the intercooler to cool the hot air delivered by the turbo compressor.

KEYWORDS: Heat exchanger core, intercooler, mass flow rate and volumetric efficiency.

1. INTRODUCTION

In turbo-charging, the volumetric efficiency of the engine is improved by pumping the air into the combustion chamber instead of sucking, as in normally aspirated engine. By doing this, volume of air sucked into the combustion chamber is more than its capacity under static conditions [1]. This is done by using a compressor driven by a turbine, both mounted on the same shaft. During this process, when the air is passed through the compressor, its temperature is increased due to compression [3]. For further reduce this temperature, the air is passed through the intercooler while this intercooler is not able to reduce the temperature of

the air as initially it was. In summers when the speed of the vehicle is low, the performance of the intercooler becomes very low. If it is possible to down the temperature of the air nearer or below the ambient temperature after intercooler and the mass flow rate will increase to a good extent. The volumetric efficiency of the engine will also increase to high extent due to increase in the mass flow rate of air, as more quantity of air will be pushed into the combustion chamber [4][5][6]. Through the review of technologies used in turbocharging, it has been concluded that intercooler is never made refrigerated to cool the hot air [2].
To achieve this, the intercooler of the turbocharger unit is made refrigerated to some extent by passing the conditioned air into it as the cold fluid, coming from cooling coil of the automotive air conditioning system. By this method, the compressed hot air will be cooled by the mixture of two, the cooled air from air conditioning system and the atmospheric air coming from the front side of the vehicle.

2. MATERIALS AND METHOD

2.1 The Detailed specifications of the engine for which the design of intercooler is being proposed

The model selected for the modification in intercooler of turbocharger is of Maruti Suzuki Swift VDi for which the technical specifications are as follows:

Table 2.1: Technical Specifications of the Engine of Maruti Suzuki Swift VDi

1.	Engine type, Valve and Fuel System		1.3L, In-line 4, 16 Valves, Dual Overhead Camshaft(DOHC), Common Rail Direct Injection (CRDi), Fixed Geometry Turbocharger with Intercooler, Diesel
2.	Displacement		1248 cc
3.	Maximum Power		76/4000 HP/rpm
4.	Maximum Torque		190/2000 Nm/rpm
5.	No. of Cylinders		4
6.	Valves per Cylinder		4
7.	Valve System		DOHC
8.	Fuel Type		Diesel
9.	Fuel System		Common Rail Direct Injection (CRDi)
10.	Drive System		2 Wheel Drive(Front)
11.	Maximum Speed		160 km/h
12.	No. of Gears		5
13.	Bore		69.6 mm
14.	Stroke		82 mm
15.	Compression Ratio		17.6:1
16.	Fuel Efficiency	Highway	19.1 Km/L
		City	14.4Km/L

2.2 Measurement of temperature and pressure of the air being supplied to the combustion chambers of the engine at specified RPMs

Data recorded by Smart Diagnostic Tool (SDT) through various sensors located at different points in the engine is as follows:

Table 2.2: Data recorded by SDT

S. No.	Parameters				
	RMP of Engine	Intake Air Temp. T_1 (^oC)	Mass Flow Rate (g/s)	Intake Absolute Pressure P_1 (KPa)	Boost Pressure P_2 (KPa)
1	1047	62	7.68	97	99
2	1171	62	8.67	97	99
3	1325	62	9.53	97	99
4	1333	62	9.58	97	99
5	1347	63	9.63	97	99
6	1570	64	11.16	97	100
7	1578	64	11.23	97	100
8	1913	64	14.03	97	102
9	2122	64	15.41	97	102
10	2397	65	17.70	97	104
11	2706	65	23.45	97	112
12	2760	65	24.53	97	115
13	2834	65	24.88	97	116
14	2891	65	26.18	97	117
15	3487	67	43.43	97	137
16	3503	67	46.18	97	142
17	3574	67	46.33	97	143
18	4377	67	58.67	97	155
19	5091	67	64.00	97	165
20	6010	67	65.60	97	165

Here, at inlet the air temperature, air pressure and mass flow rate of air after air filter (i.e. at the inlet of the compressor) and at the outlet, only the one parameter is air pressure. So, to calculate the enthalpy of air at the compressor outlet, it is required to calculate the outlet temperature. The compression in the compressor is considered as poly-tropic and for this process; the relation between temperature and pressure is as follows [8]:

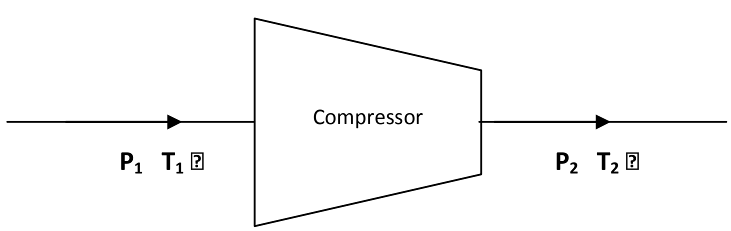

Figure 2.1: Turbocharger compressor

Let,

P_1 = Intake air absolute pressure in KPa

T_1 = Intake air tempearaure in ^oC

P_2 = Boost pressure in KPa

T_2 = Theoretical tempearaure after compressor in ^oC

T_3 = Actual tempearaure after compressor in ^oC

T_4 = Temperature of air after intercooler in ^oC

$\overset{o}{m}$ = Mass flow rate of air in g/s

n = Polytropic Index (for air, n = 1.3)

For poly-tropic process,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

Therefore, $$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

Now calculating the values of T_2 corresponding to T_1, (assuming that the compressor is frictionless) the following data has been find out [9],

Table 2.3: Temperature at the compressor outlet corresponding to intake temperature T_1

S. No.	Engine (rpm)	T_1 (^oC)	Theoretical temperature T_2 (^oC)	Actual recorded temperature T_3 (^oC)
1	1047	62	63.58	92.44
2	1171	62	63.58	92.44
3	1325	62	63.58	92.44
4	1333	62	63.58	92.44
5	1347	63	64.58	93.86
6	1570	64	66.38	94.66
7	1578	64	66.38	94.66
8	1913	64	67.93	95.32
9	2122	64	67.93	95.32
10	2397	65	70.48	98.24
11	2706	65	76.40	104.42
12	2760	65	78.54	106.25
13	2834	65	79.24	106.90
14	2891	65	79.94	106.90
15	3487	67	93.03	121.20
16	3503	67	98.25	127.25
17	3574	67	98.85	127.44
18	4377	67	105.83	135.22
19	5091	67	111.34	141.56
20	6010	67	111.34	141.56

2.3 Selection of material and design of intercooler to cool the compressed air

The compact heat exchangers are best suited for automotive applications in gas to gas heat exchange Kays and London et al. [1964]. These exchangers generally have surface area of greater than 650 m^2 per cubic meter of volume. According to the study, the suitable heat exchanger for the proposed problem should have about 800 m^2 of surface area per cubic meter of volume of cross-flow type core with both fluids unmixed. In the proposed problem, on the basis of types of fluids on both sides of the heat exchanger with both fluids unmixed and the material of heat exchanger selected, the overall heat transfer will be as 50 W/ m^2 ^oC [9]. The cold air entering in to the heat exchanger core is the mixture of air coming from automotive air conditioner cooling coil and the atmospheric air coming from the front.

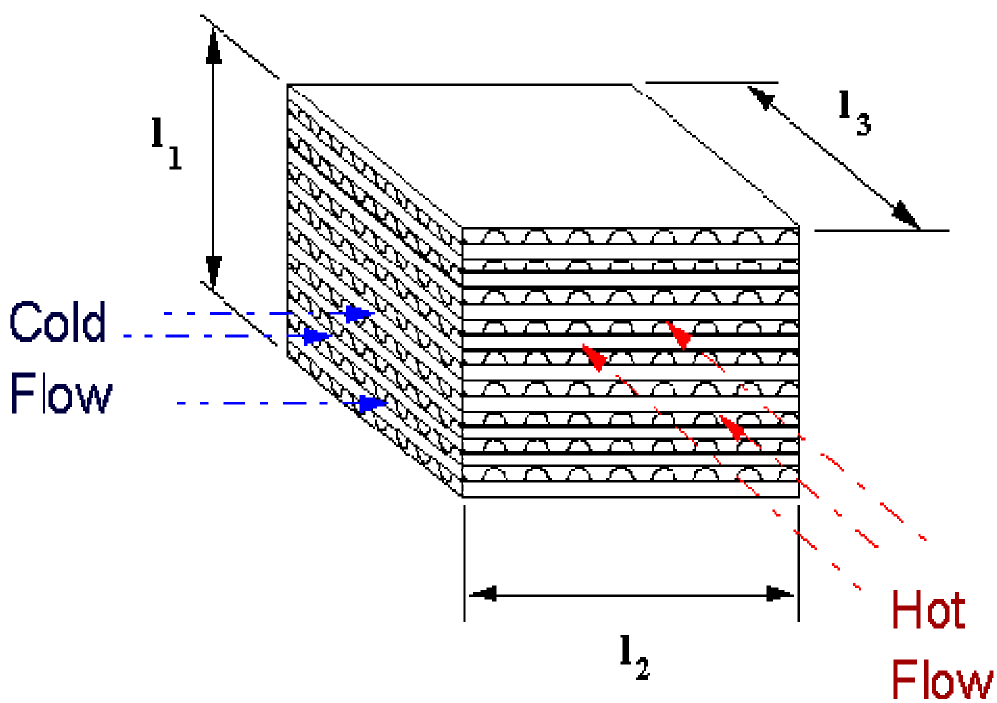

Figure 2.2: Compact heat exchanger core

3. RESULTS AND DISCUSSION

3.1 Amount of heat to be extracted from the air being supplied to the engine per second

The difference of temperature at the compressor outlet in the theoretical value and the actual recorded value (T_2 & T_3) is due to the heat added to the air because of compressor friction [4][5]. Now our purpose is to extract this heat from the compressed air before going into the intake manifold. The commercial heat exchanger used in the engine specified in the table 2.1 to extract this heat, cool the air up to a limit as recorded by the Smart Diagnostic Tool (SDT) and the data is as follows:

Table 3.1: Temperature recorded before and after intercooler

S. No.	Engine (rpm)	Temperature before intercooler T_3 (^oC)	Temperature after intercooler T_4 (^oC)
1	1047	92.44	70.25
2	1171	92.44	70.26
3	1325	92.44	70.28
4	1333	92.44	70.34
5	1347	93.86	71.42
6	1570	94.66	71.92
7	1578	94.66	72.01
8	1913	95.32	72.52
9	2122	95.32	72.61
10	2397	98.24	73.12
11	2706	104.42	74.59
12	2760	106.25	75.21
13	2834	106.90	75.32
14	2891	106.90	75.36
15	3487	121.20	77.27
16	3503	127.25	77.87
17	3574	127.44	77.91
18	4377	135.22	78.40
19	5091	141.56	79.20
20	6010	141.56	80.14

Here the purpose is to bring the temperature of air nearer to the ambient temperature by making the intercooler refrigerated. Here it is assumed to reduce the temperature (T_4) up to 20 ^oC, so the calculations for the amount of heat to be extracted per second from the air have been find out by using the following relation of heat transfer

$$\text{Heat to be extracted} = \text{mass flow rate x specific heat x (initial temp} - \text{final temp)}$$

$$= \overset{o}{m} \text{ x } C_P \text{ x } (T_3 - 20) \quad \text{J/s}$$

The specific heat of air is directly proportional to the temperature and pressure. As the temperature and pressure increases, the value of C_P for air also increases, but in the proposed problem the value of C_P is taken constant (1000 J/Kg K) throughout as the changes in the value of C_P are negligible.
Therefore, the amount of heat to be extracted at different rpm calculated by the above formula is as follows:

Table 3.2: Heat to be extracted at different RPM and mass flow rate of air

S. No.	Engine Speed (rpm)	Mass flow rate of air m_h (Kg/s)	Heat to be extracted Q (J/s)
1	1047	0.00768	556.33
2	1171	0.00867	628.05
3	1325	0.00953	690.35
4	1333	0.00958	693.97
5	1347	0.00963	711.27
6	1570	0.01116	833.20
7	1578	0.01123	838.43
8	1913	0.01403	1056.73
9	2122	0.01541	1160.68
10	2397	0.01770	1384.84
11	2706	0.02345	1979.65
12	2760	0.02453	2115.71
13	2834	0.02488	2162.07
14	2891	0.02618	2275.04
15	3487	0.04343	4395.12
16	3503	0.04618	4952.80
17	3574	0.04633	4977.69
18	4377	0.05867	6759.96
19	5091	0.06400	7779.84
20	6010	0.06560	7974.34

3.2 Calculation for the volume of heat exchanger core on the basis of heat to be extracted at different speeds of the vehicle and RPM of engine.

For designing or predicting the performance of a heat exchanger it is necessary that the total heat transfer may be related with its governing parameters:

Let,

$\overset{o}{m}$ = Mass flow rate of air, Kg/s

c_p = Specific heat of fluid at constant pressure, J/Kg ^{o}C

t = temperature of fluid, ^{o}C

Δt = temperature drop or rise of a fluid across the heat exchanger

Subscripts *h* and *c* refer to the *hot* and *cold* fluids respectively; subscripts 1 and 2 correspond to the inlet and outlet conditions respectively.

Assuming that there is no heat loss to the surroundings and the changes in potential energy and kinetic energy are negligible. From the energy balance in a heat exchanger

Heat given up by the hot fluid, $Q = m_h c_{ph}(t_{h1} - t_{h2})$

Heat picked up by the cold fluid, $Q = m_c c_{pc}(t_{c2} - t_{c1})$

Total heat transfer rate in the heat exchanger, $Q = UA\theta_m$

Where, U = Overall heat transfer coefficient between the two fluids,
A = Effective heat transfer area, and
θ_m =Logarithmic mean temperature difference (LMTD)

The expression for LMTD is essentially valid for single-pass heat exchangers. The analytical treatment of multiple pass shell and tube heat exchangers and cross-flow heat exchangers is much more difficult than single pas cases; such cases may be analyzed by using the following equation: $Q = UAF\theta_m$

Where F is the correction factor

3.3 Calculation of mass flow rate and temperature of cold air into the heat exchanger

Let,

$\overset{o}{m}_{atm}$ = mass flow rate of air coming from the atmosphere, Kg/s

$\overset{o}{m}_{aac}$ = mass flow rate of air automotive air conditioning system, kg/s

c_{patm} = specific heat of atmospheric air, J/Kg ^{o}C

c_{paac} = specific heat of automotive ac system air, J/Kg ^{o}C

t_{atm} = temperature of atmospheric air, ^{o}C

t_{aac} = temperature of automotive ac system air, ^{o}C

According to energy balance: $\overset{o}{m}_{atm}\, c_{patm}(t_{atm} - t_c) = \overset{o}{m}_{aac}\, c_{paac}(t_c - t_{aac})$

Here, the mass flow rate of cold air into the intercooler is calculated by accelerating the vehicle at constant rate in the same gear (3rd gear) so that the engine RPM and vehicle speed increases proportionally. The mass flow rate of conditioned air is considered constant. Let, the cross-sectional area of the intercooler facing front is 0.25 m^2, the density of air at normal temperature (20 ^{o}C), pressure is 1.2 kg/m^3 and the required mass flow rate of air coming from automotive AC system is 0.6 Kg/s when the duct exit area is 0.1 m^2 and the speed of air coming out of it is 5 m/s at a temperature of 15 ^{o}C, therefore the mass flow rate of mixed air (atmospheric air and the air coming from automotive AC system) entering into the intercooler can be calculated as follows:

$\overset{o}{m}_{atm}$ = speed of vehicle x cross-sectional area of intercooler x density, kg/s

Table 3.3: Mass flow rate and temperature of cold air entering into intercooler

S. No.	Engine (rpm)	Vehicle Speed (m/s)	$\overset{o}{m}_{atm}$	$\overset{o}{m}_c = \overset{o}{m}_{atm} + \overset{o}{m}_{aac}$	t_c
1	1047	0	0	0.6	15
2	1171	1.33	0.40	1.00	16.99
3	1325	4.17	1.25	1.85	18.39
4	1333	4.71	1.41	2.01	18.50
5	1347	5.32	1.60	2.20	18.61
6	1570	7.00	2.10	2.70	18.88
7	1578	7.28	2.18	2.78	18.93
8	1913	9.80	2.94	3.54	19.15
9	2122	11.2	3.36	3.96	19.24
10	2397	12.6	3.78	4.38	19.31
11	2706	14.00	4.20	4.80	19.38
12	2760	14.56	4.37	4.97	19.40
13	2834	15.40	4.62	5.22	19.42
14	2891	15.96	4.79	5.39	19.45
15	3487	18.20	5.46	6.06	19.50
16	3503	18.76	5.63	6.23	19.52
17	3574	19.60	5.88	6.48	19.54
18	4377	22.40	6.72	7.32	19.59
19	5091	24.64	7.40	8.00	19.63
20	6010	28.00	8.40	9.00	19.67

3.4 Calculation of intercooler core (heat exchanger) dimensions

The required volume of cross-flow heat exchanger core (denoted as *V*) with both fluids unmixed for different amounts of heats to be extracted at different RPMs of the engine and at different speeds of the vehicle are as follows:

Table 3.4: Required surface area of the intercooler core to reject heat

S. No	RPM	t_{h1}	t_{h2}	t_{c1}	t_{c2}	θ_m	F	Q	A	V
1	1047	92.44	20	15.00	15.92	26.22	0.976	556.33	0.4347	0.0005
2	1171	92.44	20	16.99	17.61	22.35	0.974	628.05	0.5770	0.0007
3	1325	92.44	20	18.39	18.76	18.85	0.973	690.35	0.7527	0.0009
4	1333	92.44	20	18.50	18.84	18.52	0.974	693.97	0.7694	0.0010
5	1347	93.86	20	18.61	18.93	18.44	0.974	711.27	0.7920	0.0010
6	1570	94.66	20	18.88	19.18	17.66	0.970	833.20	0.9727	0.0012
7	1578	94.66	20	18.93	19.23	17.47	0.969	838.43	0.9905	0.0012
8	1913	95.32	20	19.15	19.44	16.70	0.963	1056.73	1.3141	0.0016
9	2122	95.32	20	19.24	19.53	16.30	0.959	1160.68	1.4850	0.0019
10	2397	98.24	20	19.31	19.62	16.46	0.952	1384.84	1.7675	0.0022
11	2706	104.42	20	19.38	19.79	17.09	0.928	1979.65	2.4964	0.0031
12	2760	106.25	20	19.40	19.82	17.27	0.924	2115.71	2.6516	0.0033
13	2834	106.90	20	19.42	19.83	17.26	0.924	2162.07	2.7113	0.0034
14	2891	106.90	20	19.45	19.87	17.08	0.917	2275.04	2.9051	0.0036
15	3487	121.20	20	19.50	20.22	18.93	0.811	4395.12	5.7257	0.0072
16	3503	127.25	20	19.52	20.31	19.69	0.761	4952.80	6.6107	0.0083
17	3574	127.44	20	19.54	20.31	19.57	0.753	4977.69	6.7557	0.0084
18	4377	135.22	20	19.59	20.51	20.29	0.749	6759.96	8.8963	0.0111
19	5091	141.56	20	19.63	20.60	20.83	0.747	7779.84	9.9997	0.0125
20	6010	141.56	20	19.67	20.55	20.44	0.746	7974.34	10.459	0.0131

From the results shown in the table 3.4, it has been concluded that the maximum volume of the core of heat exchanger required is 0.0131 m^3, rejecting the heat at maximum speed of engine. As it has already been discussed in the previous section that the required cross-sectional area of the heat exchanger is 0.25 m^2, so the required dimensions of the intercooler core will be as **length (l_1) =0.5 m, breadth (l_3) =0.5 m and thickness (l_2) =0.0524 m.**

4. CONCLUSION

When the normal intercooler is used, the temperature of air in the intake manifold ranges from 70-80 ^oC in the pressure range of 99-165 KPa and in this range the density of air ranges between 1.0053 Kg/m^3 to 1.6281 Kg/m^3. By adopting the above specified dimensions of the intercooler (heat exchanger) core, it will be possible to supply the air to the intake manifold at the temperature 20 ^oC. With the application of the designed dimensions of the intercooler core, the temperature in the intake manifold will remain 20 ^oC and the pressure ranges from 99-165 KPa and in this range of temperature and pressure, the density of air will range from 1.1769-1.9614 Kg/m^3. The result shows that there is high gain in the mass flow rate of air in to the engine on using the above designed intercooler. Due to this more oxygen will be available in the combustion chamber to burn the fuel. Increasing the oxygen content with the air leads to faster burn rates and the ability to control exhaust emissions. Added oxygen in the combustion air offers more potential for burning fuel.

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