Analytical Analysis of Electric Vehicle Chassis Frame and Battery Thermal Management System

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Abstract. The energy stored in the battery is the source of the energy to drive the electric vehicles. At the moment the size and the weight of the battery pack required for given mileage are very much high when compared to its counterpart IC engine. The main aim of this work is to modify the existing electric vehicle chassis frame for giving the provision of swapping battery rather than having permanent battery pack on the chassis frame. Specifically by taking RAV4 V5 EV as a baseline and all analysis was done on it, the static, dynamic and static impact (crush analysis) on finite element (ANSYS) is performed, battery thermal management system in the transient analysis by using EXCEL and ANSYS FLUENT is identified, and finally, prototype fabrication is carried out to demonstrate the shape of the chassis frame. According to study, the Material AISI 1020 and rectangular cross-section in 80 mm × 40 mm × 4mm is selected to the electric vehicle chassis frame and the modelling was done by using CATIA V5 and the analysis using ANSYS. Finally, the total deformation of chassis frame is 1.007 mm, equivalent stress is 115.88 MPa and the life of the chassis on full load is 130 days without deformation. The weight of the electric vehicle battery pack is decreased from 383.51 kg to 345.615 kg by modification of existing cooling plate and the chassis frame has a longer life than previous design because of the weight minimization of the battery pack on the existing vehicle.

Keywords: Batteries pack · Chassis frame · Electric vehicle · Swapping and BTMS

1 Introduction

In the last decades, with the increasing development of battery technologies and concerns for the environment, electric vehicles (EVs) technologies got rapid development. However, EV drivers have to face the problem of battery refuelling daily. Once the battery runs out, drivers can recharge it in the charging station. The battery takes 6–8 h to full charge in AC charging according to the manufacturer specified time. That means waiting a long time to refuel the battery in a permanently attached battery pack is as the main problem. And this study is going to develop swappable battery pack the electric vehicle

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and detail analysis on the frame study, battery thermal management system (BTMs) and weight minimization on the battery pack as case study on a baseline of RAV4 EV battery model, this existing vehicle uses permanently attached battery pack. As the main aim of this work is to design and modify of the existing RAV4 EV 2014 model electric vehicle [1] chassis frame for swapping mechanism of battery packs and check the thermal management system of the battery pack after design the chassis frame.

The following studies are separately done and taking this separate study as a gap and in this work going to merge those studies of the chassis frame and electric vehicle battery pack. The selection of structural steel AISI 1020 is done on the study of [2].

The battery and thus also the battery frame does not need to be considered in the weight limit of 400 kg for the empty vehicle [3]. Therefore, it is obvious to use the structure of the battery frame also for other functions, besides carrying the battery modules. The weight of the frame is still relevant because increased vehicle weight results in decreased vehicle performance and increased energy consumption. Although a frame made from steel profiles will be heavier than a frame made from aluminium extrusion profiles, it was decided to make the frame from steel because of cost reasons. The other one is Range anxiety [4] is a relatively new concept which is defined as the fear of running out of power when driving an electric vehicle [6–8]. To decrease range anxiety result that they got were that with bigger batteries this is still not the answer for the anxiety. So, swapping the battery pack is the best way to minimize this anxiety according to this work [9–11].

The other big issue in the electric vehicle battery pack is battery thermal management system (BTMS) [5] for optimal performance, safety and durability considerations, the battery must operate within the safe operating range of voltage, current and temperature as indicated by the battery manufacturer [12–14]. The voltage range is between the maximum voltage (3.65 V) and the minimum voltage (2 V), the temperature range is depending on the operating mode (charge and discharge) and varies between [−20 °C; 60 °C] and [0 °C; 40 °C] during discharge and charge, respectively. These ranges can change according to the cell chemistry and battery manufacturer. By adding an extra feature on study this work takes the temperature of Adama, Ethiopia and the analysis is done on locally [15, 16].

2 Material and Methods

2.1 Materials for Electric Vehicle Chassis Frame

The material AISI 1020 is selected for this electric vehicle chassis frame according to the literature discussion, the availability, cost and its mechanical properties that suit for chassis frame design and the rectangular cross-section was taken in reference of literature on related studies.

2.2 Steel AISI 1020 the Properties from ANSYS 15 Workbench

That the preferred material for this electric vehicle chassis frame is steel AISI 1020 and its mechanical properties is listed in Table 1 below.

Table 1. Table for material properties

Properties of structural steel AISI 1020 which data's from ANSYS	
Compressive yield strength MPa	250
Tensile yield strength MPa	250
Tensile ultimate strength MPa	460
Reference temperature °C	22
Relative permeability	10000

2.3 Specification for RAV4 EV 2014 Model

By taking this permanently attached battery pack of RAV4 EV 2014 model electric vehicle into swappable one to afford alternative rather than wasting time to charge the battery (by swapping the battery pack). The specification mentioned in Table 2 below. All the study is done on this specification.

Table 2. Specification for RAV4 EV 2014 model

Mechanical and performance	RAV4 EV specification
Motor type (two drive modes; normal and sport)	AC induction motor with fixed gear open-differential transaxle
Power output	154 hp (115 kw) max
Max. torque	218 lb, -ft/273 lb, -ft
Max. speed	85 mph/100 mph (160 kph)
Type	Lithium-ion (Li-ion)
Power output	400 W max.
Voltage	386 V max.
Weights and capacities	
Curb weight (lb)	4032
Seating capacity	5
Cargo volume (cu. Ft.) behind front/second row	73.0/36.4
Gross vehicle weight rating (GVWR) (lb.)	5005
Battery capacity (kwh)	35.0/41.8
Battery weight (lb.)	845.5
EPA-rated driving range	103 miles
Charging time (normal/extended charge mode)	12 A/120 V, 44 h/52 h 16 A/240 V. 12 h/15 h 30 A/240 V. 6.5 h/8 h

2.4 Design, Modeling and Analysis of Chassis Frame

The software *CATIA* for model drawing, *ANSYS* for finite element analysis and *EXCEL* for transient analysis for battery thermal management system is used.

Methodology

The following Fig. 1 shows the working flow for this research work.

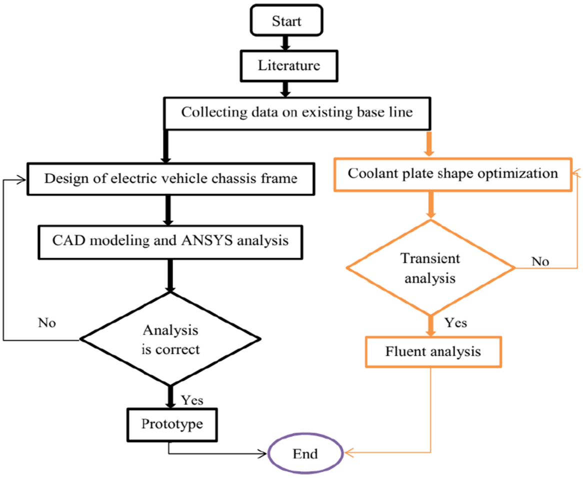

Fig. 1. Flow chart for how this work is done

Prototype Fabrication for Electric Vehicle Chassis

The prototype Fig. 2 is only for the demonstration 1:1.5 scale of the actual one.

Fig. 2. Chassis frame for designed electric vehicle

3 Design, Modeling and Analysis of the Chassis Frame

The chassis generally experiences four major loading situations; vertical bending, longitudinal torsion, lateral bending, and horizontal lozenging. Understanding these conditions is the key to designing a better chassis. When the chassis frame supported at its ends by the wheel axles and acted upon by an equivalent weight due to the vehicle's equipment, passengers and luggage around the middle of its wheelbase, the side members are caused to sag in the central region. This sagging is known as vertical bending [6].

General calculations for a considered load on baseline chassis frame is.

- Gross vehicle weight = 2270.22 kg
- Total load to be applied = 2270.22 × 9.81 = 22,270.86 N
- Considering an overload of 25% of the total load = 22,270.86 × 1.25 = 27,838.57 N.
- Chassis has two beams. So load acting on each beam is half of the Total load acting on the chassis. Load acting on the single frame = 27,838.57 N./2 = 13,919.285 N/Beam

a. The calculation for the frame

The uniformly distributed load is 13,919.285 N/4,524 mm = 3.07 N/mm taking the reaction support A. as shown in Fig. 3.

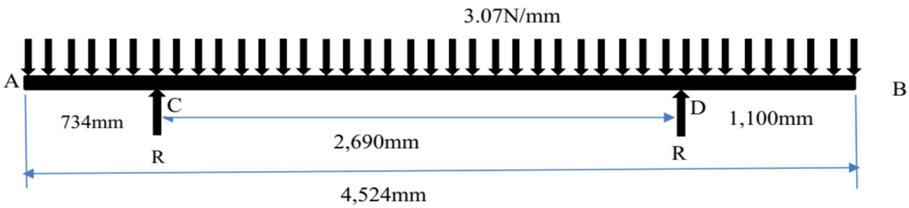

Fig. 3. Chassis as a simply supported beam with overhung

$$Rc + RD = 13,919.285\,\text{N} \quad (1)$$

$$RC = \frac{Wl(l - 2c)}{2b}\ [7] \quad (2)$$

$$Rc = 5,999.5\,\text{N}$$

$$RD = 7,919.8\,\text{N}$$

Where,

$Rc\ \&\ R_D = reaction\,forces\,at\,point\,C\ \&\ D$
$W\ = distributed\,load\,over\,the\,member,$

$l = length\,of\;the\,side\,member,$
$(a, b, c) = lengths\,between\,reaction\,support\,and\,edge$

Mathematical analysis for shear force and bending moment
Shear force

$V_A = Wa = 2,815.19\,\mathrm{N}$
$V_C = RC - V_A = 3,184.31\,\mathrm{N}$
$V_B = WC = 3,377\,\mathrm{N}$

$$V_D = RD - V_B = 4,542.8\,\mathrm{N}$$

Where: V is shear force,
Bending moment

$$M_{left} = -\frac{wa^2}{2} = -1{,}290{,}764.615\,\mathrm{Nmm}$$

$$M_{right} = -\frac{wc^2}{2} = -1{,}857{,}350\,\mathrm{Nmm}$$

$$M_{midd} = Rc\left(\left(\frac{Rc}{2w}\right) - a\right)$$

$$= 1{,}458{,}582.024\,\mathrm{Nmm}$$

The shear force diagram for electric vehicle chassis frame based on the above calculations Fig. 4 for shear force diagram and Fig. 5 stands for bending moment.

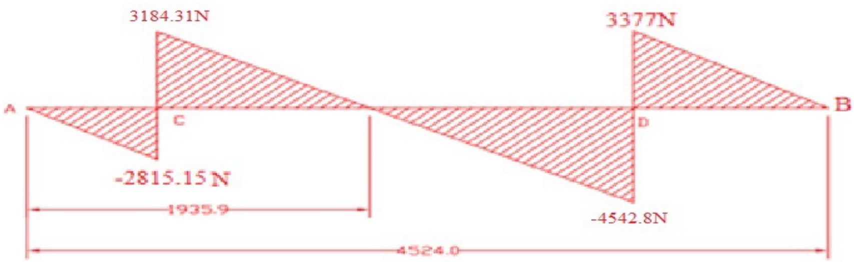

Fig. 4. Shear force diagram

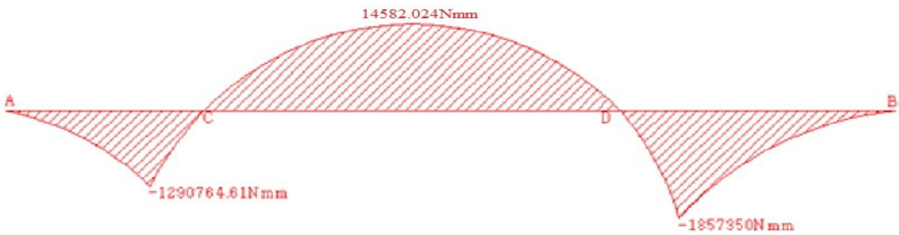

Fig. 5. Bending moment diagram for frame

From the above Fig. 4 and Fig. 5 taking that the distance or position of the shear stress occurs on the chassis frame and the position of bending moment. According to this reason position for both shear force and bending moment lies on the position 1935.9 mm distance from the front end of the chassis frame so this value is used to find the deflection (Y) in the following equation.

The calculation for stress generated

$$M_{max} = 1{,}458{,}582.024\,\text{Nmm}$$

Moment of inertia around the x-x axis in the following Fig. 6.

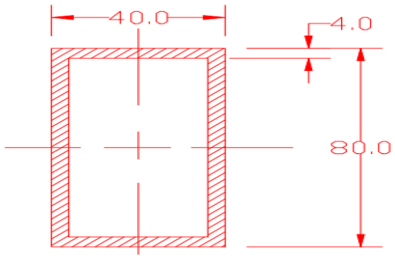

Fig. 6. Cross section for the chassis frame

$$I_{xx} = \frac{bh^3 - b_1{h_1}^3}{12} = 711{,}338.667\,\text{mm}^4$$

Deflection of the chassis frame

$$Y = \frac{wx(b-x)}{24EI}[x(b-x) + b^2 - 2\left(c^2 + a^2\right) - \frac{2}{b}c^2x + a^2(b-x) = 0.87\,\text{mm}$$

The deflection of the total chassis frame is 0.87 mm from its normal position and the stress on the chassis frame is done in the following manner.

$$\frac{\sigma}{Y} = \frac{M}{I} = \frac{E}{R}$$

Where ‘M’ is bending moment, ‘Y’ is the distance between the xx section and the normal stress centerline; ‘I’ is mass moment of inertia in xx cross-section

$$\sigma = 104.62\,\text{MPa}$$

CAD Modelling for Electric Vehicle Chassis

The chassis frame model created in CATIA V5 R20 as shown in Fig. 7 is imported to ANSYS 15. The meshed model of chassis frame is shown in Fig. 9 (Fig. 8).

Fig. 7. Designed electric vehicle chassis

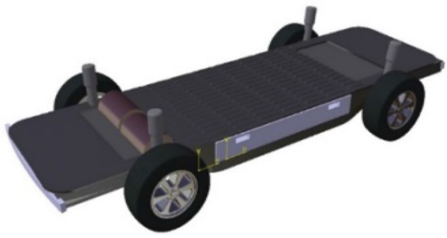

Fig. 8. Designed 3D model

Meshing. The meshing of the model of chassis frame 243634 Nodes and 34528 elements.

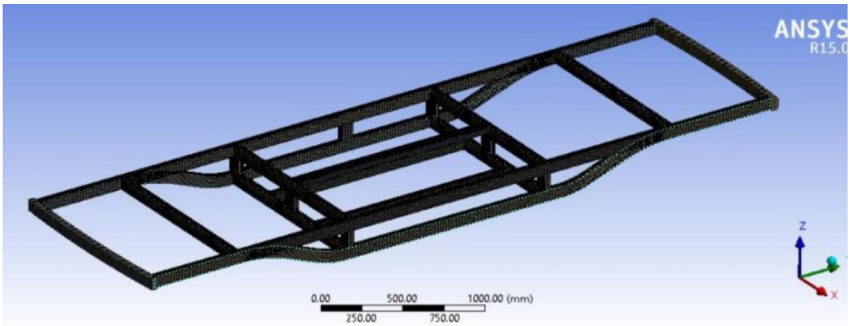

Fig. 9. Meshed model

Loading and Boundary Conditions

In this RAV4 EV 2014 model vehicle taking that of the existing vehicle load and the analysis is held by this load. The magnitude of the force on the upper side of chassis is 117720 N which is carried by the distributed load on chassis frame on Fig. 10.

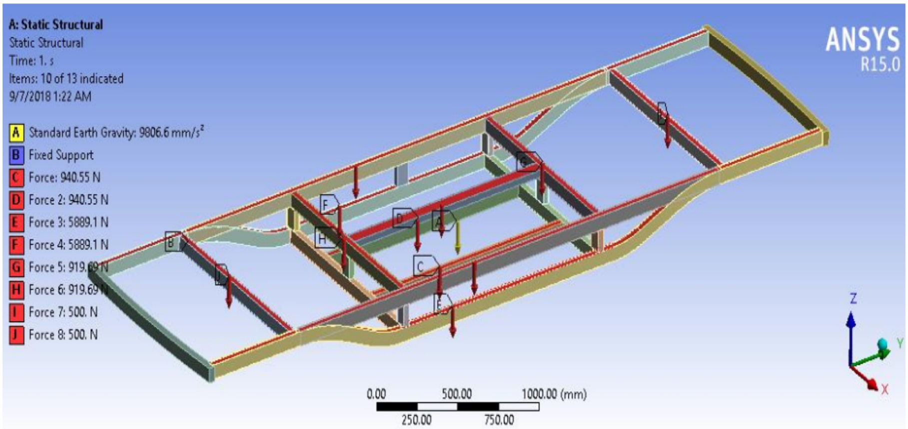

Fig. 10. Static loads (loads from the battery, passenger and luggage) on the chassis frame

b. Battery thermal management system (BTMS)

Shape Minimization of the Cooling Plate

The existing battery cell on rav4 EV is pouch type and its arrangement by two layers 7 batteries (170 mm) in vehicle track position and 7 batteries (position of 265 mm) in parallel to vehicle base and 7 batteries are out of normal arrangement then the total number the battery cell is 105. The mass of the single battery cell is 1.4285 kg and the mass of the cooling plate in one side of the battery cell before minimization is 1.1178 kg. Figure 11 shows that the noted scheme in visual.

Transient Analysis for the BTMS

The derived differential equations, those represent the energy balance for the battery cell cooling plate, coolant plate and fluid equations are solved using the explicit finite difference method. The time derivative is replaced by the forward difference scheme. The transient temperatures are evaluated iteratively, using relations derived for the equation of energy balance [8]. The diameter of the copper coil and naming its top part of the plate (p1) and bottom plate (p2) respectively in Fig. 12, below.

For p1 (Top Part of the Plate)

$$E_{in} - E_{out} = \Delta E \tag{3}$$

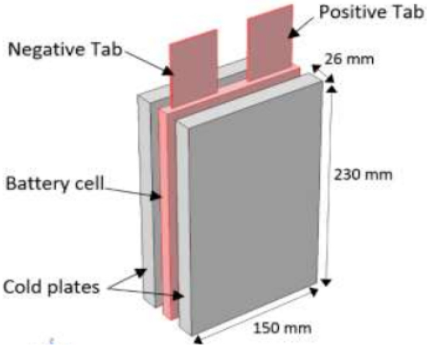

Fig. 11. Battery cell with its cooling plate

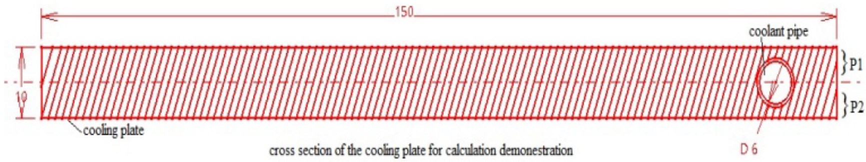

Fig. 12. Cross-section of the aluminium plate

$$heat\ coming\ from\ battery - heat\ transfer\ to\ fluid = \rho_p V_p C_p \frac{dT_{p1}}{dt}$$

By discretizing the above energy balance to get the following equation for the Sect. 1 or the above plate

$$heat\ generated - h_{fluid} A\left(T_p - T_f\right) = \rho_p V_p C_p \frac{dT_{p1}}{dt} \tag{4}$$

For Fluid

$$-\dot{m}_f C_f \frac{\partial T_f}{\partial x} + h_f \pi D\left(T_{p1} - T_f\right) - h_f \pi D\left(T_f - T_{p2}\right) = \rho_f A_f C_f \frac{\partial T_f}{\partial t} \tag{5}$$

For p2 (Bottom Part of the Plate)

$$h_f A\left(T_f - T_{p2}\right) - h_{air} A_s\left(T_{p2} - T_{air}\right) = \rho_{p2} V_{p2} C_{p2} \frac{dT_{p2}}{dt} \tag{6}$$

The explicit finite-difference method is proposed to solve the system of equations. The time derivatives are replaced by a forward difference scheme, whereas the dimensional derivative is replaced by a backward difference scheme.

For p1 (Top Part of the Plate)

$$E_g - h_f A\left(T_p^t - T_f^t\right) = \rho_p V_p C_p (T_f^{t+\Delta t} - T_f^t) \tag{7}$$

For Fluid

$$-\dot{m}_f C_f \frac{\left(T_{fj}^t - T_{fj-1}^t\right)}{\Delta l} + h_f \pi D\left(T_{p1j}^t - T_{fj}^t\right) - h_f \pi D\left(T_{fj}^t - T_{p2j}^t\right) = \rho_f A_f C_f \left(\frac{T_{fj}^{t+\Delta t} - T_{fj}^t}{\Delta l}\right) \tag{8}$$

For p2 (Bottom Part of the Plate)

$$\rho_{p2} V_{p2} C_{p2} \left(\frac{T_{p2}^{t+\Delta t} - T_{p1}^t}{\Delta l}\right) = h_f A\left(T_f^t - T_{p2}^t\right) - h_{air} A_s (T_{p2}^t - T_{air}^t) \tag{9}$$

From the EXCEL transient analysis, the following figures founded it indicates that the temperature variation on the cooling plate in each minute. Some assumptions for the analysis are taken out.

These are

- The average speed of the vehicle 90 km/hr (25 m/s)
- The average temperature of the environment 25 °C in Adama city
- Assume that the driver drives the vehicle for 5 h (300 min) without rest
- Taking the elbow of the copper coil as the node, then the analysis has nine nodes

ρ_f(kg/m^3)	= 997	k_f(w/m.k)	= 0.607
ρ_{air}(kg/m^3)	= 1.225	h_f	= 257.32
ρ_p(kg/m^3)	= 2700	E_g(kw)	= 645
C_f (J/kg. k)	= 4185.5	T_{air}	= 25
C_p	= 921	D_p	= 0.007
h_{air} (W/m^2 k)	= 52.09	L(m)	= 1.4
P_{rf}	= 6.14	μ_f	= 0.000891
f_{inlet}	= 20 °C	$T_{initial}$	= 25 °C

Constants and calculated values for the EXCEL input are as follow.

The values like Reynolds number, the volume of plate and volume pipe, volume of fluid and area are calculated values the constants took from thermodynamics table of cengle in 25 °C.

The Nusselt, Rayleigh, and Prandtl numbers are given by [8]:

$$R_l = \frac{\rho_{air} V_{air} L}{\mu} \tag{10}$$

$$\boldsymbol{R_l = 3.682 \times 10^5}$$

$$P_r \, from \, thrmal \, property \, tables \, and \, charts$$

$$N_u = 0.56 R_l^{1/2} P_r \tag{11}$$

$$\boldsymbol{N_u = 469.66}$$

Where

$R_l = Renould\ number$
$P_r = Prandtl\ number$
$N_u = Nusselt\ number.$

CAD Modelling to the Cooling Plate

After performing the transient analysis by EXCEL, CAD modelling by CATIA v5 software and the analysis by ANSYS 15 for the fluent (CFD) is processed below in Fig. 13.

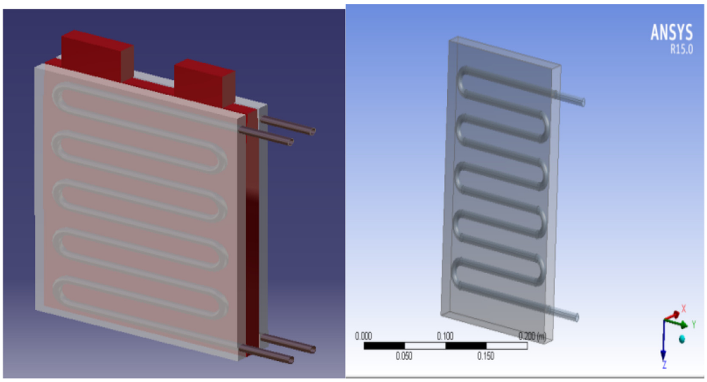

Fig. 13. Cooling plate with the battery cell

The maximum temperature registered in Fig. 14 is 319.2 K when it changed to degree Celsius it becomes

$$C^o = K - 273$$

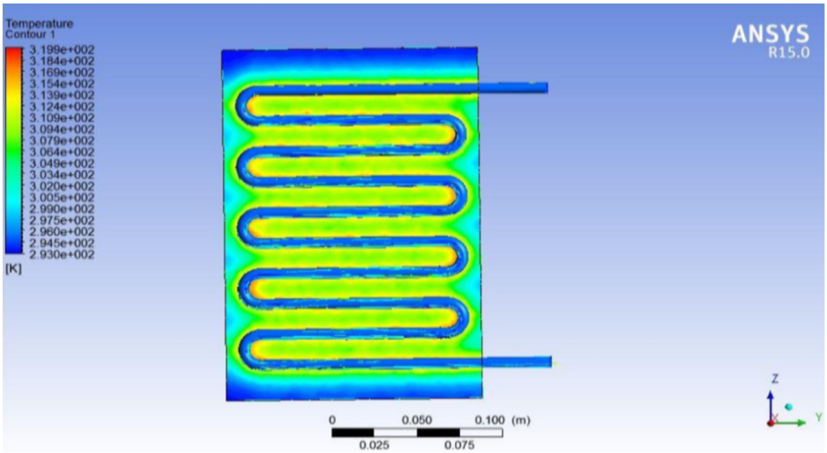

Fig. 14. Temperature contour result for the new cooling plate sectioned view

$$C^o = 319.2 - 273$$

$$C^o = 46$$

According to the excel result, we can see the effect of the fluent result on the cooling plate of the temperature of the entire plate with the coolant fluid temperature is 46.2 °C as shown on the above CFD analysis in figure Fig. 14. So, it is validated.

4 Result and Discussion

a. Static structural analysis for electric vehicle chassis frame

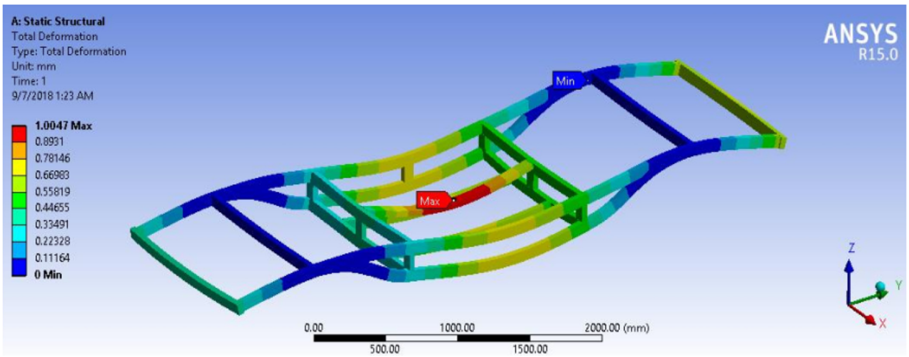

Fig. 15. ANSYS static result of total deformation

In the above figure Fig. 15, we understand that from the static structural analysis the total deformation is 1.0047 mm it is the maximum deformation in this loading.

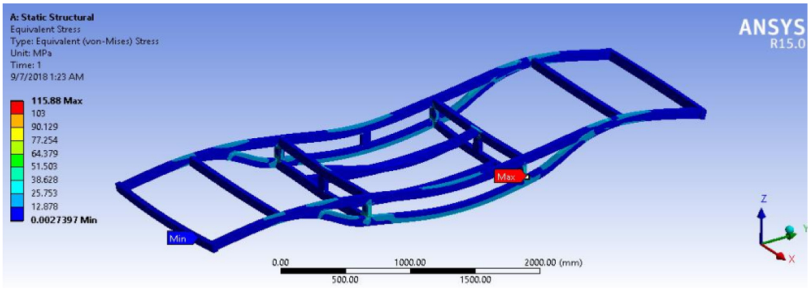

Fig. 16. Static structural analyses for equivalent stress (von-misses stress)

In the above figure, Fig. 16, the static structural analysis for equivalent stress (von-misses stress) is 115.88 MPa is the maximum stress and the minimum von-miss stress 0.00273 MPa.

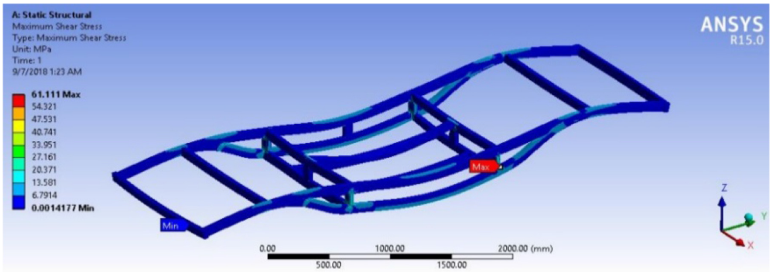

Fig. 17. Static analysis of maximum shear stress

Figure 17, express that the static structural analysis of shear stress on the chassis frame and the maximum shear stress is 61.111 MPa and the minimum shear stress is 0.001417 MPa.

Fatigue Life. The loading is of constant amplitude, this represents the number of cycle's1million cycle until the part will fail due to fatigue and the safety factor for this analysis is 0.743. The given load history represents 188,480 min of loading with this load condition and the chassis will fall after this minute this means that it stays by this loading for 130 days without deformation.

Fatigue Sensitivity. Fatigue Sensitivity shows how the fatigue results change as a function of the loading at the critical location on the model. This result may be scoped. Sensitivity may be found for life, damage, or factor of safety. And in the following ANSYS result based on the applied loading up to 72%, the frame has the same available life cycle after that when the percentage of loading is more the available life cycle decreased. The value only 72% is that much not good but based on the considering safety factor of chassis frame is 0.743 this safe and the taken personal Wight as 75 kg. Default values for the sensitivity options may be set through the Control Panel.

b. Result for static impact analysis

The following Table 3 shows that results from the finite analysis.

Table 3. Result for static impact analysis

	Force applied (N)	Max. stress (Mpa)	Max. deformation (mm)	Factor of safety
Front	19900	456.59	3.207	0.1889
Rear	8613	359.59	1.8599	0.2397
Side	8613	192.79	2.317	0.447

c. Result for battery thermal management system

Figure 18 shows that the average temperature on plate facing on the battery cell (P1) of nine nodes in 300 min then the temperature is 38 °C to 47 °C smoothly increasing in each minute, the average fluid temperature of ten nodes is between 20 °C and 29.5 °C and the upper part of a plate (P2) average temperature is 21 °C to 28.08 °C. The correct mass flow rate for gating this temperature is the mass flow rate of 0.001 kg of the fluid. And the following graph describes the relations and features of the three case temperatures in nine nodes and 300 min.

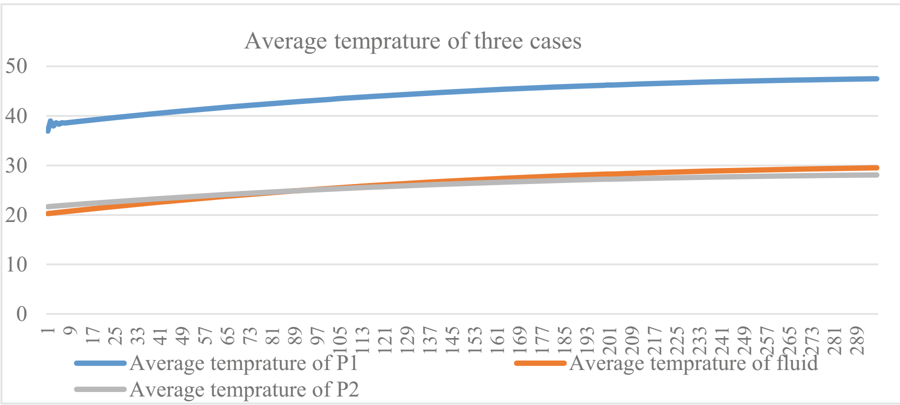

Fig. 18. Average temperature of p_1, fluid and p_2

5 Conclusions

The structural analysis includes the total deformation of the chassis frame on ANSYS 15 and the analytical work of the result. So, the total deformation on the ANSYS 15 workbench is 1.0047mm. And the von-misses stress is 115.55 MPa on the ANSYS result it shows the chassis is in safe mode. The maximum shear stress on the ANSYS is 61.111 MPa occurs on the right-rear part of the battery set. The fatigue life of the designed chassis frame is 130 days with a full load without deformation and sensitivity of the electric vehicle chassis frame able to 72% additional weight in 10e6 cycle beyond this percentage the sensitivity cycle tends to decrease.

That what is done in the BTMS is shape minimization of the cooling materials like a plate and cooper coil of the coolant the previous size of the cooling plate is 230 mm × 150 mm × 12 mm and the diameter of the coil is 9 mm and after shape minimization, the size changed to 230 mm × 150 mm × 10 mm and the copper coil diameter is 7 mm. the need of this minimization is to reduce the weight of the battery pack from 383.51 kg to 345.61 kg there is no problem happened on the entire structure of the battery because the transient analysis is done and the shape concerned with only the cooling plate, not the battery itself.

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