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Cfd analysis of HVAC triangular duct for rail compartment

Mohammed Nomaam[1], md Ismail[2], Ather ahmed[3], Arfath khan[4], mohd Khaleel ullah [5], mohd Moizuddin[6],syed Rahman [7],Ruhail masood[8]
[1,2,3,4,5,6,,7,8](mechanical engineering jntuh), india
Corresponding Author: Mohammed Nomaam

ABSTRACT : *The hvac system is not just limited to the convectional calculation and layout but as the systems are getting complex new designs have to be developed that meet the criteria in consideration of efficiency, price, overall system requirement. One of the implementations is in the railway requirement. Which is a challenging consideration for two criteria's one being the space and other is the weight.so the convectional branching method is hard and the compartment consists of a single zone.It becomes more complex because the convectional ducting system layout becomes hard due to space restriction. The study made in this paper is divided into two parts. first is to replace the complex ducting layout with simplex ducting system both in design and manufacturing by triangular layout. And second is to validate the triangular duct design with cfd for two parameters one is only supply but no suction pressure and the next is same supply but some negative suction pressure. The cfd results are evaluated using the high-fidelity lattice Boltzmann method (LBM) algorithm using ANSYS solver.comparing both the cases the best one is selected for the conjugate heat transfers analysis. The heat generation is simulated considering of 4 solids each generation heat flow of 1495 watt.*
Keywords : *cfd, hvac, lbm, ansys*

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I. INTRODUCTION

Hvac system elaborates itself as heating ventilation and air conditioning. The system when implemented for the rail compartment becomes more complex due to less space and have a variational load based on the humans sudden entering the zone. For the even distribution of air, the ducting system is implemented that carries the air from the either side of the package unit into the compartment and then delivered by the diffusers as designed by the convectional method. The paper tries to validate a simple duct system which could replace the convectional duct layout both in complexity and easy to manufacturing.

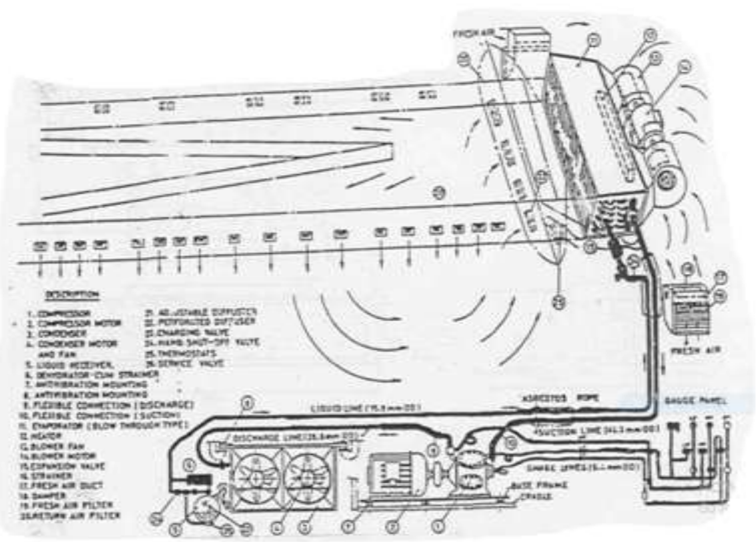

Fig.1. shows the old Ac unit system mounted in ICF coaches.

Fig.2. shows the new package Ac unit mounted on the top of compartment in LHB coach

II. THERMAL LOAD AND COOLING CALCULATION

Designing for any hvac system for railway the main factor of consideration is the heat load, temperature, passenger load. The standards for the design have been taken from theRDSO specification no TRC-1-72 like air required for passenger per min, heat transfers for various material in the compartment

T.D. for end portions is always considered to be 3°C less than T.D. for other parts of the coach, since non-airconditioned space adjacent to the airconditioned compartments is considered to have a temperature of 3°C less than the ambient temperature.

Requirement of fresh air for non-smoking= 0.35m^3 /passenger/ minute.
compartments
Quantity of ventilating air for 46 passengers 0.35x46 = 16.1 m^3 / minute 16.1x35.3= 568.33 (CFM)

The following are the wattages considered for various-electrical appliances.
2 Flourescent tube light - 24 W.

Eventhough the wattage of the tube is 20 W, the choke also consumes energy. Hence,
1.2 times the wattage i.e. 1+2 x 20 = 24W has been considered for the purpose of heat load calculations.

Incandescent lamps
Carriage Fan =
= 15W
29W

DATA COLLECTED FROM A.C. MANUAL

Heat transfer from equipment's and fans	=	2545 BTU/HP/Hr
Heat transfer from fluorescent lightsand incandescent lamps	=	3.4 BTU/Watt/Hr.
Sensible heat per passenger	=	205 BTU/Hr. (51.6 K.Cal/Hr)
Latent heat per passenger	=	195 BTU/Hr (49.12K.CaI/Hr)
1 Ton of refrigeration	=	12000 BTU/Hr. (3024 K.Cal/Hr)
1 k-calorie	=	3.97 BTU/Hr.

Coefficient of heat transfers for some materials

Wall end partition	=	0.615
Roof	=	0.65
Floor	=	0.72
Window	=	1.92
U for window	=	5.34

The calculation is based on full load of 46 passengers with all the heat source running, including electrical etc.

DIMENSIONS OF A.C. PORTION OF COACH -

Length of AC portion (A)	=	15.2 M
Width of roof (B)	=	3.245 M
Width of floor (C)	=	3.04 M
Height of A.C. portion (D)	=	2.03 M
Area of side wall (A x D)	=	30.856 M^2
Area of roof (A x B)	=	49.324 M^2
Area of floor (A x C)	=	46.208 M^2

Area of end partitions Height of window	=	6.17 M^2
Width of window	=	0.56M
	=	0.61M

Area of window 0.56 X 0.61	=	0.3416 M^2
No. of windows per side wall	=	16
Total area of windows per side wall	=	0.56x0.61x16=5.466M^2
Area of side wall excluding windows	=	30.856 - 5.466 = 25.2 39M^2.

CONNECTED ELECTRICAL LOADS INSIDE A.C. COMPARTMENT

Fluorescent lights 2' long	=	30 Nos.
Incandescent lamps	=	16 Nos.
Fans	=	8 Nos.
Blower Fan motors (0.65 HP)	=	2 Nos.

1. Heat gain due to conduction = AxKxTDX3.97BTU/Hr.

Sidewall: 50.78 x 0.615 x 20 x 3.97 = 2479.64 BTU/Hr. (624.59 K.Cal/Hr.)

Roof	:	49.324 x 0.65 x 20x3.97	=	2545.61 BTU/Hr. (641.21 K.Cal/Hr.)
Floor	:	46.208 x 0.72 x 20x3.97	=	2641.62BTU/Hr. (665.4 K.Cal/Hr.)
End partition	:	2 x 6.17 x O.615 x (20 – 3) 1 7x3.97	=	512.288 BTU/Hr.
Window	:	5.466 x 2 x 1.94x20 x 3.97	=	1683.8 BTU/Hr.
Total	:	2479.64 + 2545.61 + 2641.62 + 512.288 + 1683.8	=	9862.954 BTU/Hr.…(I)

2. Solar Heat Gain: A x K x TDS x3.97

Sidewall: 25.39 x 0.615 x 9x3.97 = 557.92BTU/Hr.
(140.53 K.Cal/Hr.)

Roof : 49.324 x 065 x 10.55 x3.97 = 1342.81 BTU/Hr
(338.24 K.Cal/Hr)

Window : 5.466 x 5.34 x 95.55x 3.97 = 11071.34 BTU/Hr.
(2788.75 K.Cal/Hr)

Total : 557.92 + 1342.81+
11071.34= 12972.069 BTU/HR…(II)

3. Heat gain due to passengers (BTU/Hr.)

S.H.	=	205 x No. of passengers.	
L.H.	=	195 x No. of passengers.	
S.H + L.H	=	400 X No. of passengers.	
	=	400 x 46 = 18400 BTU/Hr.	…(III)
	=	(4634.76 K.Cal/Hr)	

4. Heat gain due to ventilation (BTU/Hr.)

S.H.	=	1.08 x Q x TD x 9/5	
	=	1.08 x 568.33 x 20 x 9/5	
	=	22096.67 BTU/Hr.	
	=	(5565.91 K.Cal./Hr)	
L.H.	=	0.68 x Q x Gd	
	=	0.68 x 568.33 x 26	
	=	10048.07	
	=	(2531 K.Cal./Hr)	
Total	=	22096.67 + 10048.07	
	=	32144.7 BTU/Hr	… (IV)
	=	(8096.91 K.Cal./Hr)	

5.

.Heat gain due to elect, appliances

	=	Wattagex3.4 BTU/Hr. or,
		H.P. x 3600 BTU/Hr.
Flouroscent Light 20W	=	(20 x 1.2) W.
	=	1.2 x 20 x 3.40 x 30
	=	2448 BTU/Hr.
	=	(616.62 K.Cal./Hr)
Incandescent lamps	=	15 x 16 x 3.40
	=	816 BTU/Hr.
	=	(205.54 K.Cal/Hr)
Fan	=	29W x 8 x 3.4
	=	788.8 BTU/Hr.
	=	(198.69 K.Cal./Hr)
Blower fan	=	0.65HP x 2 x 2545
	=	3308.5 BTU/Hr
	=	833.37 K.Cal./Hr ...(v)

Total =2448+816+788.8+3308.5=(7361.3 BTU/Hr)=1854.22 K.Cal/ Hr

Total of l + II + III + IV+V = 80741 023 BTU/Hr (20337.78K.Cal/Hr)

Heat gain due to infiltration@ 10% = 8074.1 BTU/Hr.
= (2033.78 K.Cal./Hr)

Gross TotalHeatgain = 81003 07 -8100.3
= 88815BTU/Hr.
= 22371.56 K.Cal./Hr

Refrigeration capacity (TR) = $\frac{88815}{12000}$ =7.4 TR(for one package unit)

III. CAD DESIGN AND MODEL TESTING

The first part deals with the cad model generation of the LHB coach. Then for the triangular duct the outlet volume is calculated. Such that the new triangular design meets the standards of delivering equal cfm in every diffuser. The duct is designed using the latest space claim direct modeler software and then transferred to the cfd.Results are evaluated using the high-fidelity lattice Boltzmann method (LBM) algorithm using ANSYS solver. For the first case of duct the duct is supplied with 3000 cfm on each side and the outlet pressure is maintained at 0 Mpa. For the second case the supply is same but the outlet suction pressure is maintained at negative 2 Mpa to compare the results of efficient system.

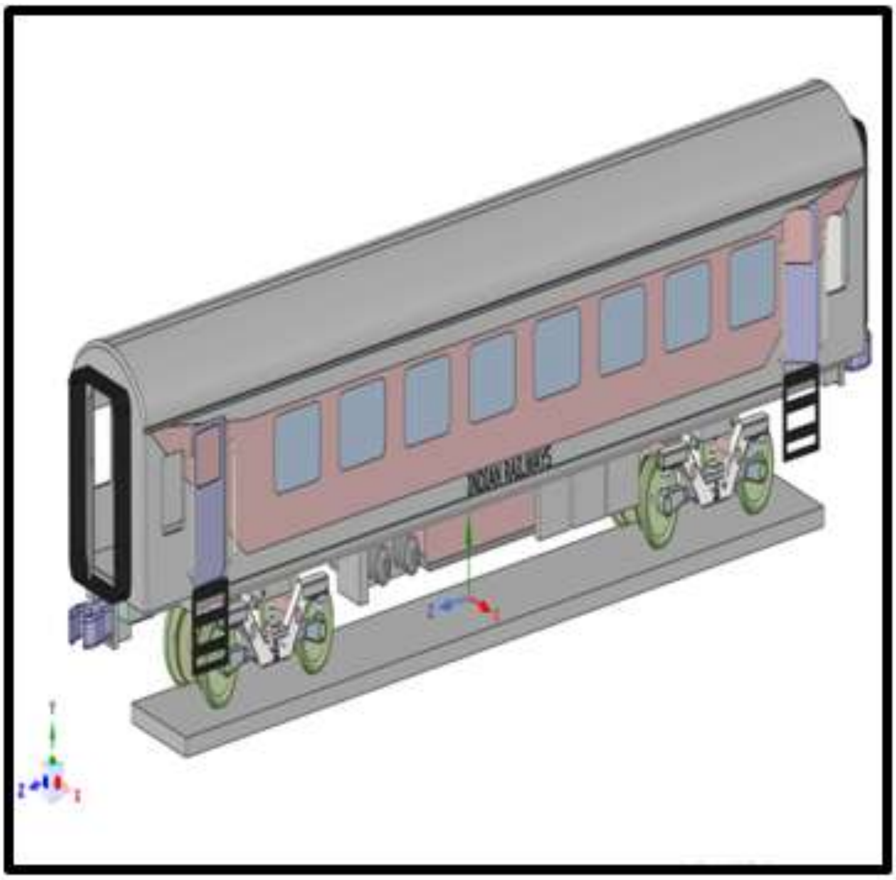

Fig. 3 shows the CAD model generation of LHB coach in space claim design modeler

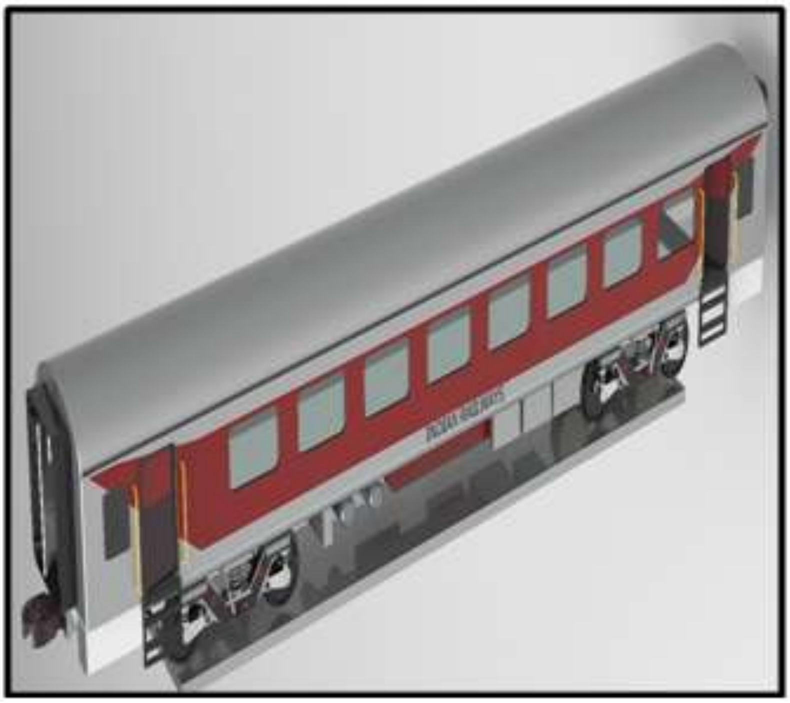

Fig. 4 shows the rendered model generation of LHB coach.

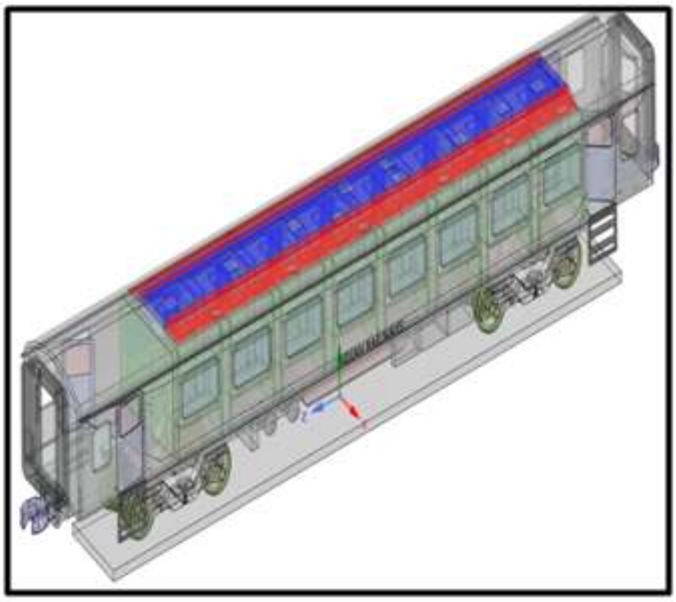

Fig. 5 shows the convectional supplyduct (blue) and returnduct (red) duct of LHB coach.

The total cooling load supply estimated as per the RDSO specification no TRC-1-72 for LHB coach is 7.4 tons for each package unit so now the duct size is calculated and the [width/height>4] ratio is taken into consideration. So, this 7.4 tons for each unit there are two package units on either ends so the total load is 14.8 tons. That equals to a flow rate of 6000 cfm. And the duct size is calculated as shown below.

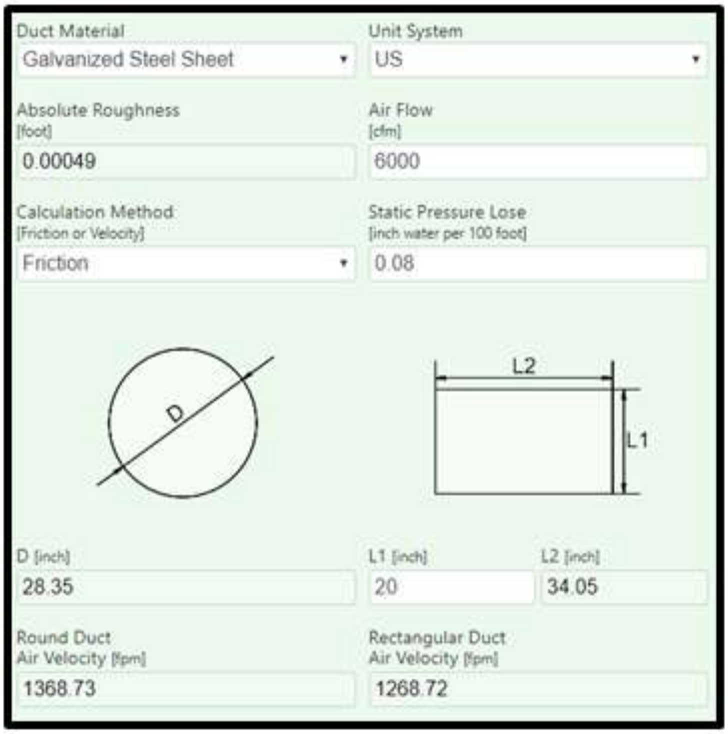

Fig. 6 shows the duct size calculated 20 inch * 34-inch, L/D ration of 1.7

The next part is to design the triangular duct based on the above dimensions and the calculated cooling load of 6000 cfm. the rectangular duct is designed in the space claim software and split diagonally forming the two triangular split ducts. the main aim here is to see the flow rate is maintained constant for each diffuser as in a convectional duct. The advantage of this duct over the convectional duct is easy onsite manufacturing, less complexity, even distribution of air. The initial supply of air is 3000 CFM on each end, so a total of 6000 cfm. The figure below shows the further process.

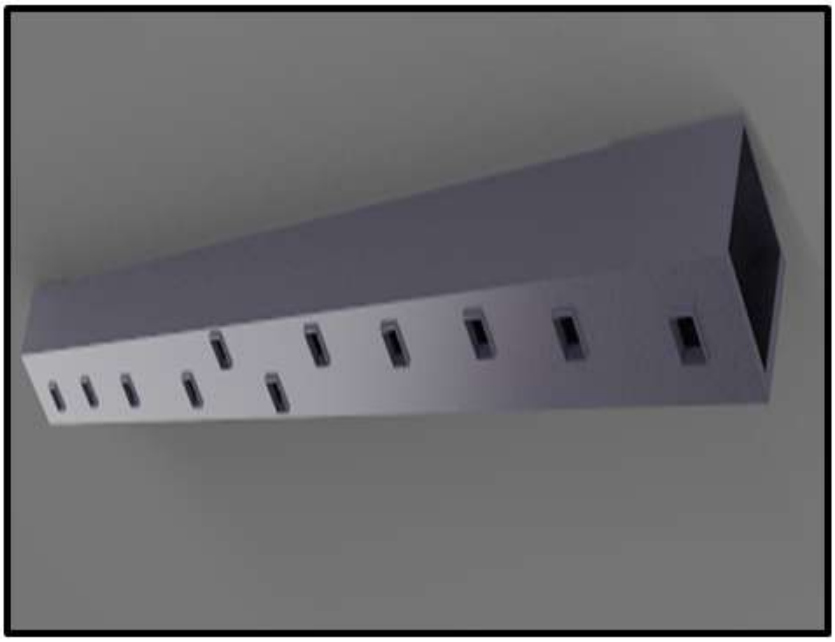

Fig. 7 shows the internally modified duct design the splitting into two triangles.

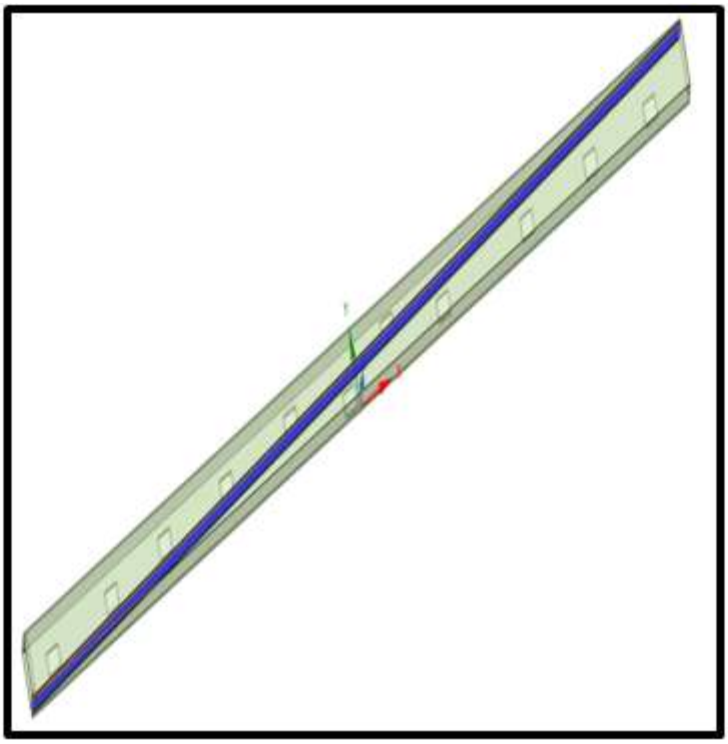

Fig. 8 shows the internally modified duct design the splitting by blue wall into two triangles.

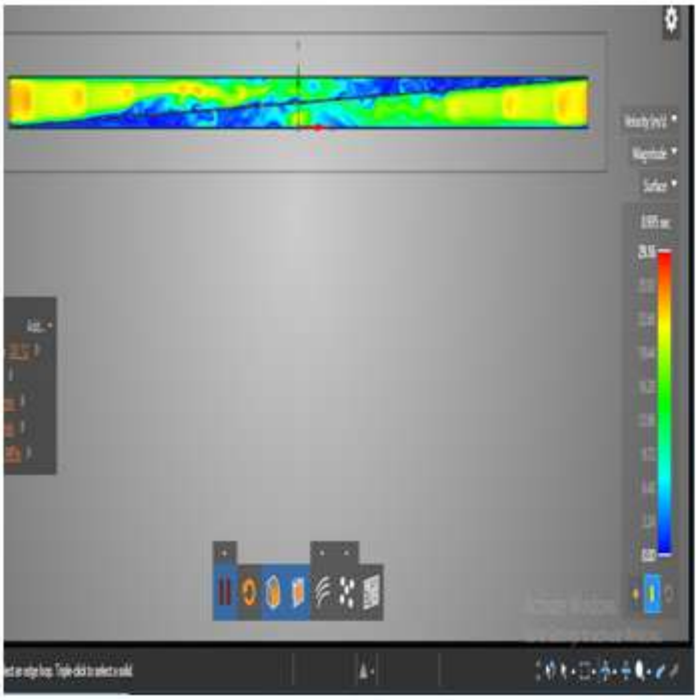

Fig. 9 shows the internally modified duct velocity distribution

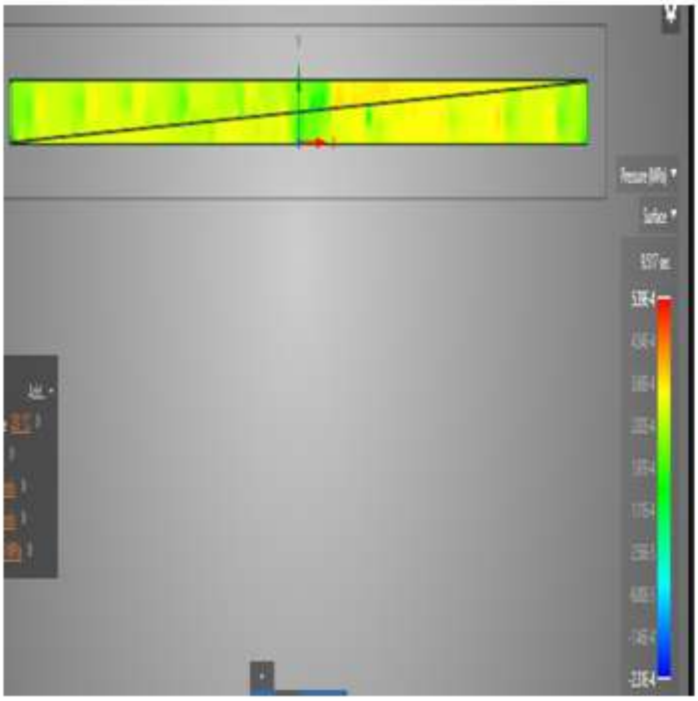

Fig. 10 shows the internally modified duct pressure distribution

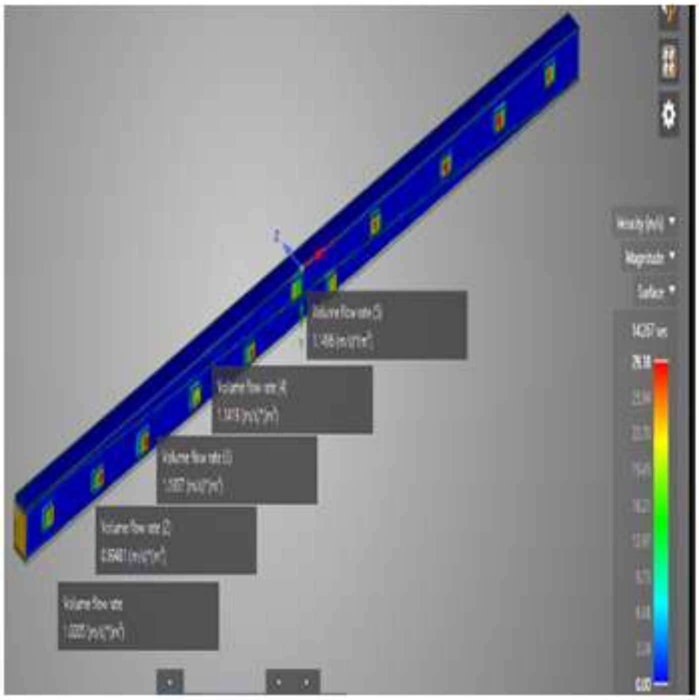

Fig. 11 shows the internally modified duct volume distribution

Diffuser no.	Flow rate (m/sec) *m^2
1	1.0205
2	0.99481
3	1.1957
4	1.1419

Table. 1 shows the volumetric flow rate being constant at every diffuser in the optimized design

The next part is to assign this triangular duct to the rail compartment and to observe the flow pattern and for the two cases as follows. The rectangular inlets and circular outlets are shown

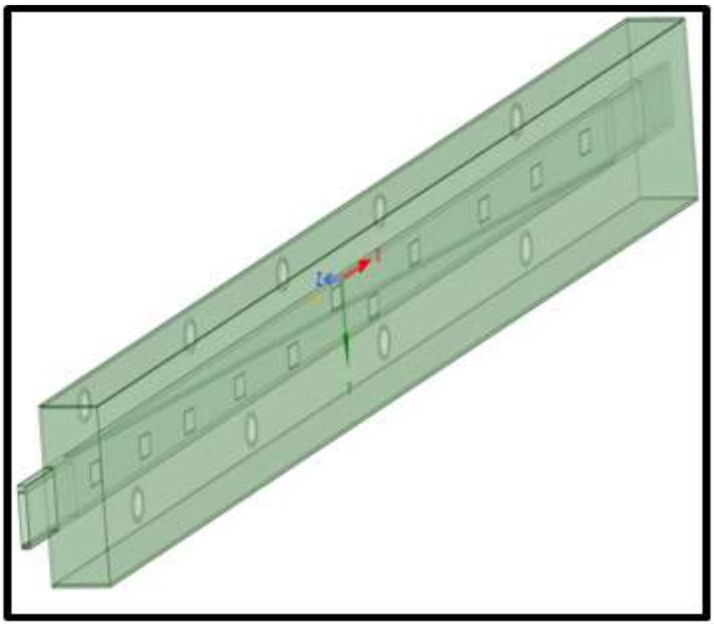

Fig. 12 shows the triangular duct is placed inside the compartment having rectangular supply and circular return pattern. And the fluid domain is extracted

Case -1: the system is supplied with 3000 CFM on each side of the duct and the outlet is maintained at 0 Mpa.

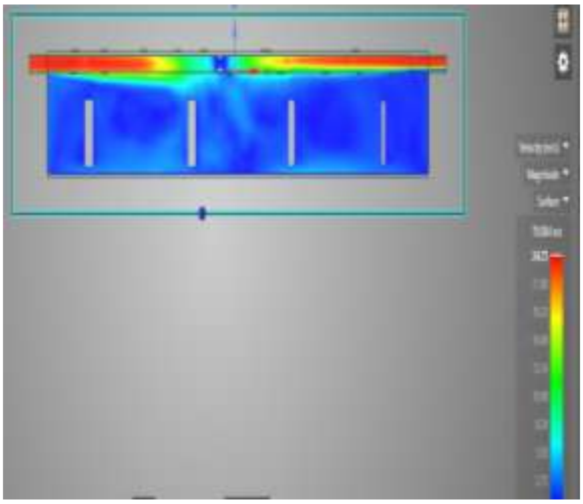

Fig. 13 shows the duct velocity distribution for supply 3000 cfm and return 0 Mpa.

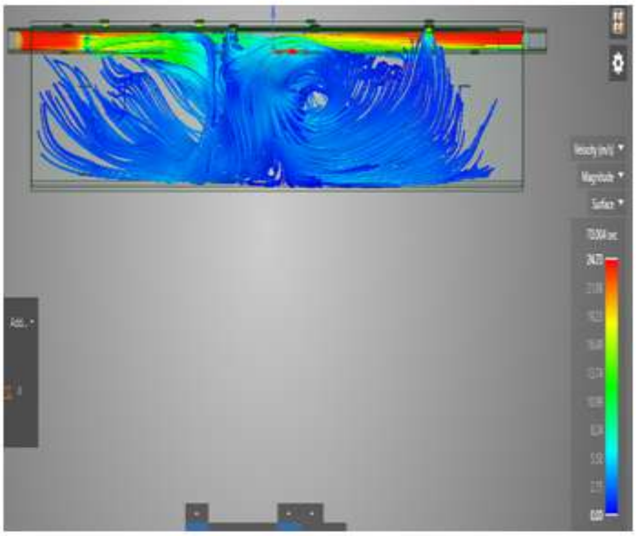

Fig. 14 shows the duct velocity streamlines for supply 3000 cfm and return 0 Mpa.

Case -2: the system is supplied with 3000 CFM on each side of the duct and the outlet is maintained at -2Mpa.

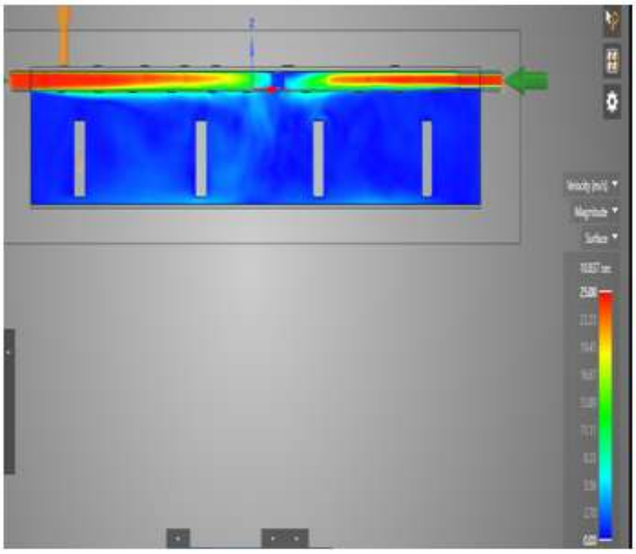

Fig. 15 shows the duct velocity distribution for supply 3000 cfm and return -2Mpa.

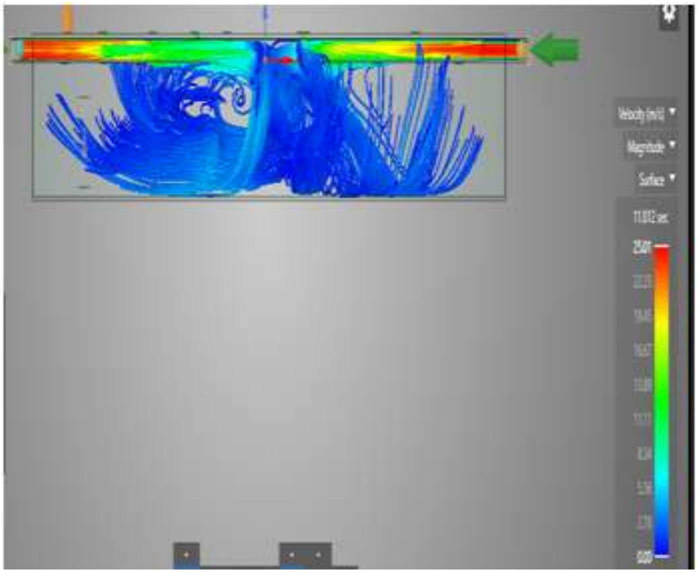

Fig. 16 shows the duct velocity streamlines for supply 3000 cfm and return -2 Mpa.

The first case and second case were compared it was found that the second case has good turbulence of air due to the negative suction pressure of -2 Mpa. So, second case was selected for the conjugate heat transfers. The general heat flow per person is 130 Watt. So, we assume to have 46 passengers in compartment.

130watt * 46 passengers = 5980 watt (total)
For analysis complexity this heat flow of 5980 watt is divided into four solids of each 1495 watt.

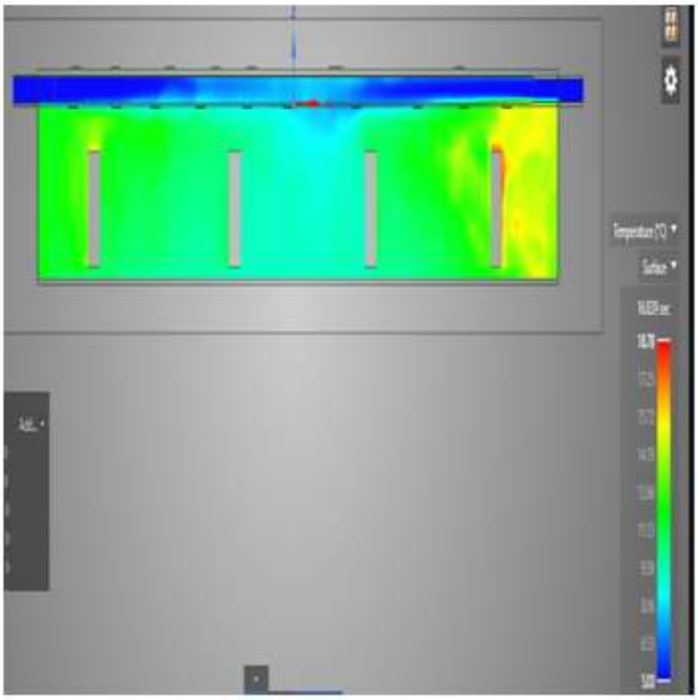

Fig. 17 shows the thermal distribution of air temperature. ranging from 5 degree C of supply to max 18.78 degree C in the compartment.

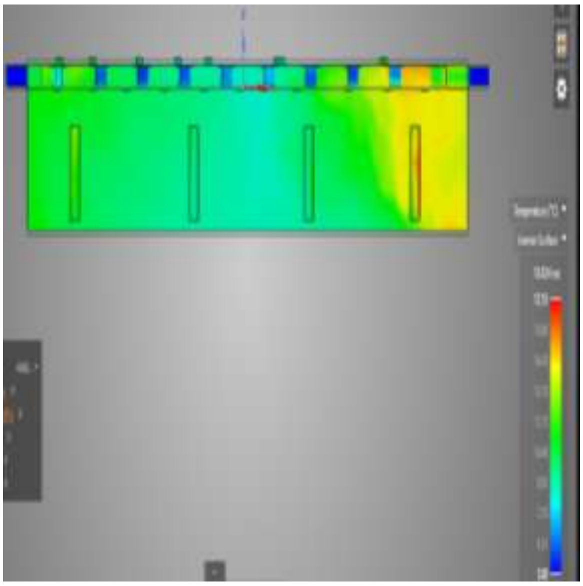

Fig. 18 shows the thermal distribution over the four solid bodies.

IV. CONCLUSION

Considering the climatic changes around. there is necessity of air conditioning throughout the year. Various applications ranging from residential, commercial buildings to moving vehicles is seen. The hvac system in rail compartment becomes more complex due to less space and light weight demand. So innovative ways should be adopted through CFD in order to make the system efficient. The paper tries to compare the result for two cases of the new triangular duct considering for an equal distribution of air through every diffuser. It's found that the second case having a total flow of 6000 cfm and suction pressure of – 2 Mpa was found to be efficient as the air is circulated evenly. the cfd results are evaluated using the high-fidelity lattice Boltzmann method (LBM) algorithm using ANSYS solver. Triangular duct construction when compared to the general ductand branching is more efficient for compact spaces.

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