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The Impact of Tube Optimization Design on Radiator Heat Rejection Performance

Toure Ismael and Buyun Sheng
Department of Mechanical and Electronic Engineering, Wuhan University of Technology, Wuhan, China (NIGER)

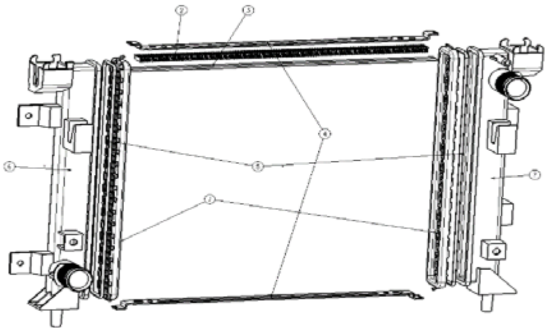
***Abstract*—This work consists to evaluate the impact of the tube design on the car radiator heat rejection performance. For this work we come up with two propositions, the first one consists to change the traditional tube shape "1.6x16 B-type" to "1.6x16 Dot B- type". The second one is just to change the traditional tube dimension from 1.6x16 to 1.4x16. After the two samples radiators were made, we will proceed to their performance analysis such as the heat rejection, the coolant pressure drops, and the airside pressure drop. For these performance tests we will use the company laboratory Multi-functional test bench (T-HWS-2H), and some mathematical formula to calculate our samples performance. At last we will compare our analysis results with the radiator "1.6x16 B-type" heat rejection data (Req.), which structural model and data is our research basic standard. The results show that the tube design has great impact on the radiator heat rejection performance.**

This work has been performed at HUBEI RADIATECH COOLING SYSTEM CO., LDT (HBR).

Keywords- car radiator; tube type; fin

I. INTRODUCTION

The air-cooled heat exchangers found in a vehicle (radiator, AC condenser and evaporator, charge air cooler, etc.) has an important role in its weight and also in the design of its front-end module, which also has a strong impact on the car aerodynamic behavior. The radiator is the main component in a cooling system to remove heat from the engines. Looking at these challenges, an optimization process is compulsory to obtain the best design compromise between performance, size/ shape and weight. This experience's objective demands advanced design tools that can indicate not only the best solution but also the fundamental reason of a performance improvement that will satisfy our customers and market demand. To improve the heat transfer from the surface, we use the B shape tubes which are vital components in the radiator design, the liquid flows in B shape tubes while the air flows in channels set up by multileveled fin surfaces. In many situations, the thermal resistance on the air side is larger than that on the liquid side. Lu et al. [1-2] also adopted the porous medium model to simulate the pressure loss through the horizontal radiator in a small dry cooling tower. [3-4] analyzed the performance of a spiral tube heat exchanger, in comparison with shell and tube heat exchanger. Their optimized spiral design revealed that, heat transfer is enhanced compared to the shell and tube heat exchanger. Different experimental studies were performed to analyze and verify their advantages in various heat exchange systems like shell and tube heat exchangers [5], double tube heat exchangers [6-7], plate heat exchangers [8], heat pipes [9-10], microchannel heat sink [11], electronics cooling [12], building air conditioning [13], and the like. Leong et al. [14] attempted to investigate the heat transfer characteristics of an automotive car radiator using ethylene glycol based copper nanofluids numerically Ismael,T et al [15] presented a set of parametric studies of heat dissipation performed on automotive radiators by of designing five radiators with different fin pitch wave distance (P = 2.5, 2.4, 2.3, 2.2, 2.1 mm).



7	Right tank	1	198	PA66-GF30(Zytel® 70G30HSLR BK099)/2.5mm	/
6	Left tank	1	207	PA66-GF30(Zytel® 70G30HSLR BK099)/2.5mm	/
5	Gasket	2	13	EPDM(E6272)/3mmx4mm	/
4	Sider	2	35	HF8535/O/1.5mm Core alloy:HF342(AA3003MOD.) Braze alloy:HF401(AA4343)	/
3	Tube	37	284	HF8181/H24/0.23mm Core alloy:HF329(AA3005Mod) Braze alloy:HF403(AA4343) Waterside alloy:HF335(AA3003Mod)	/
2	Fin	38	155	HF306/H16/0.06mm*16mm Core alloy:HF306(AA3003Mod)	/
1	Header	2	164	HF9771/O/1.5mm Core alloy:HF350(AA3003Mod.) Braze alloy:HF401(AA4343) Waterside alloy:HF333(AA3003MOD)	/
NO.	Name	Quantity	Weight (g)	Material	Remarks

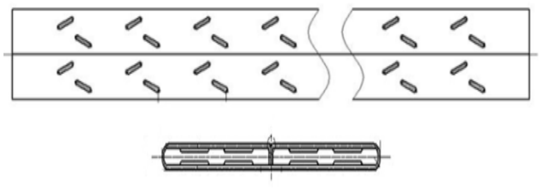
FIGURE I. RADIATOR STRUCTURAL DESIGN AND NOMENCLATURE

II. EXPERIENCE PROCEDURE AND RESULTS

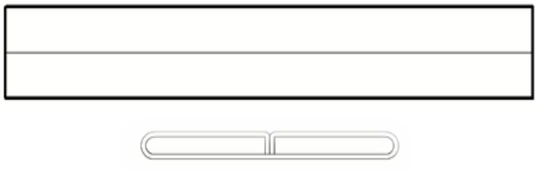
The analysis focuses on the cooling performance for automobile radiator by changing the tube design and parameters. For the cooling performance experience, we use T-HWS-2H Multi-functional test bench for Automobile and Tractor Radiators. The test bench system is a continuous air suction type wind tunnel in a chamber which can also control the ambient air temperature; collection and control of operating condition parameters can be done automatically by the computer via the preset program, and also can be done by the user manually. We also use some mathematical formula according to the heat dispassion factors we develop in the company.

A. *First Proposition*

We use CATIA V5-6R2014 to draw our designed sample, which consist to change the tube type from “1.6x16 B-type” to “1.6x16 B-Dot type” as shown in figure 2. After the new sample is made we precede to his analysis true the wind tunnel in the company laboratory.



1. 6x16 Dot type



1.6x16 B-type

FIGURE II. TUBE STRUCTURAL DESIGNS

ANALYSIS RESULTS

TABLE I. WIND TUNNEL BENCH ANALYSIS RESULTS DATA

Pth (KW)		T coolant=90° C						
		1200L/h	**2400L/h**	**3600L/h**	**4800L/h**	**7200L/h**	**8400L/h**	**Pext (Pa) @1200 L/h**
Tamb= 20° C	**1m/s**	**7.4**	**7.7**	**7.8**	**7.8**	**7.8**	**7.8**	**6.5**
	2m/s	**12.5**	**14.0**	**14.7**	**15.1**	**15.5**	**15.5**	**25.7**
	3.5m/s	**16.9**	**20.5**	**22.7**	**24.2**	**25.9**	**26.3**	**68.1**
	5m/s	**20.0**	**24.5**	**27.3**	**29.5**	**32.8**	**34.0**	**107.2**
	6.5m/s	**22.1**	**27.1**	**29.4**	**31.2**	**34.3**	**35.6**	**157.6**
	8m/s	**23.5**	**29.9**	**33.5**	**36.6**	**41.7**	**43.9**	**209.4**
	△ Pint(mbar) @5 m/s	**40.0**	**154.5**	**348.0**	**618.9**	**1393.2**	**1896.7**	

TABLE II. REQ AND HBR DATA COMPARISON

Operating point	**Air speed [m/s]**	**Tamb [° C]**	**Coolant Flow [L/min]**	**T inlet [° C]**	**Heat Exchange [Kw]**	**△Pext Max [Pa]**	△ Pint Max [Kpa]
Req.	**2.00**	**20**	**16**	**90**	**11.20**	**28.20**	10.85
HBR		**20**	**16**	**90**	**11.70**	**25.60**	2.62
Req.	**5.00**	**20**	**16**	**90**	**17.00**	**116.60**	10.85
HBR		**20**	**16**	**90**	**17.90**	**106.30**	2.68

- The impact of B-Dot type shape tube on the heat rejection parameters

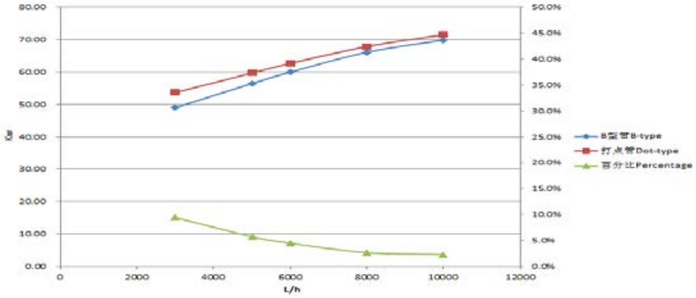


FIGURE III. HEAT REJECTION

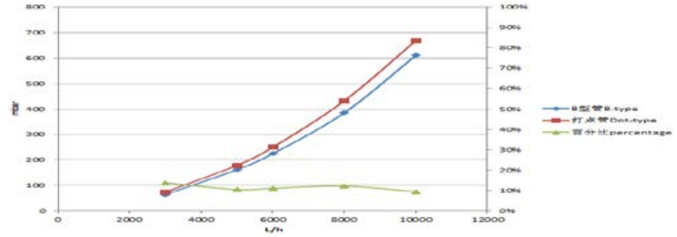


FIGURE IV. COOLANT PRESSURE DROP

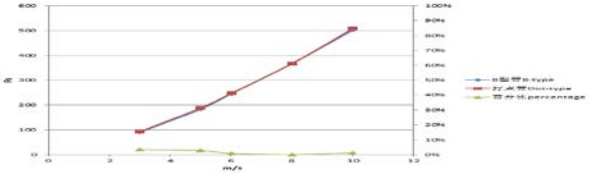


FIGURE V. AIRSIDE PRESSURE DROP

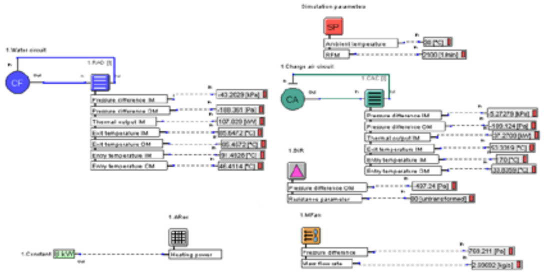


FIGURE VI. KULI PERFORMANCE SIMULATION

According to our research and analysis results, after we changed the tube B-shape to Dot B-type, it has great impact on the radiator working condition. From fig 3 to fig 5 we can observe that the radiator heat rejection increases about 5%, coolant pressure drops about 15%, and the airside pressure drop remain almost invariant. As we know from the radiator working condition the smaller the flow rate is, the higher the heat rejection will increase. That`s means the Dot B-shape tube reaches our goal which is to increase the radiator heat dissipation performance, but it`s also improve the tube structural strength. From our analysis experience data on the radiator and other products we`ve work on in the past, we know that if the heat rejection decreases by 1%, airside pressure drop will decrease by 3%. According to this relation between the two factors, if we decrease the heat rejection by 5%, the airside pressure drop will also decrease by 15%, as shown in the performance data from tab 3 to tab 4. After decreasing the heat rejection by 5%, the sample we design meet the requirements, but still have one point who is still higher (yellow part in tab 4) than the standard sample data (Req.). But that is not a problem because the radiator we use in this work has a small flow rate, that`s means the heat rejection improvement may exceed 5%. So, there will be more space to adjust the airside pressure drop. In the future we will make some new samples to measure the performance value, and then make some adjustment.

TABLE III. REQ AND HBR DATA COMPARISON (1.6X16 B-TYPE)

Radiator Heat Rejection (KW)		Air Velocity m/sec (rt =70 deg)				Coolant Pressure Drop (Kpa)	
		2 m/s		5 m/s			
		Req.	HBR	Req.	HBR	Req.	HBR
Coolant Flow	16L/Min	10.5	11.0			10.85	2.85
	16L/Min			14.7	17.0	10.85	2.85
Airside Pressure Drop (pa)		26	36.3	110.0	126.7		

Heat rejection 5%
Airside pressure drop 15%

TABLE IV. REQ AND HBR DATA COMPARISON (1. 6X16 DOT TYPE)

Radiator Heat Rejection (KW)		Air Velocity m/sec (rt =70 deg)				Coolant Pressure Drop (Kpa)	
		2 m/s		5 m/s			
		Req.	HBR	Req.	HBR	Req.	HBR
Coolant Flow	16L/Min	10.5	10.5			10.85	2.85
	16L/Min			14.7	16.3	10.85	2.85
Airside Pressure Drop (pa)		26	30.9	110.0	107.7		

B. *Second Proposition*

This solution consists to change our standard prototype tube size from 1.6x16 to 1.4x16, and then run some heat rejection performance analysis true mathematical formula and the laboratory heat rejection test bench.

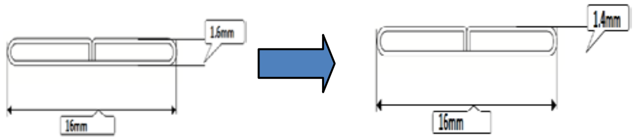
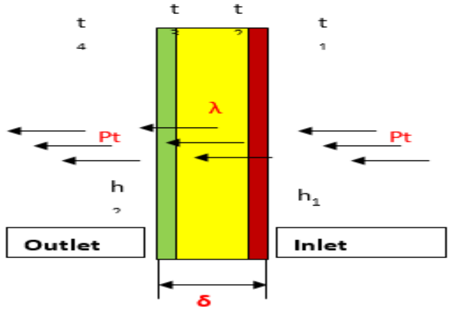


FIGURE VII. B SHAPE TUBE FROM 1.6x16 to 1.4x16

Coolant Flow	Coolant Pressure Drop
[L/min]	[Kpa]
16	2.48

- Heat rejection analysis

Pth=K*A*ΔT
Pth ——Heat rejection
K ——Transfer coefficient
A ——Surface area
ΔT ——Temperature difference
th =1/ (1/h1/A1+ δ/λ/A1+1/h2/A2) *(t1-t4)
h1——Internal Convection coefficient
h2——External Convection coefficient
A1——Internal surface area
A2——External surface area
λ ——Heat conductivity coefficient
δ ——Tube thickness



There are three principal parameters, K, A and ΔT that affect the radiator heat rejection. In general, ΔT is almost invariant, so we only need to analyze K and A parameters.

When the core size is constant, the tube height changed from 1.6 to 1.4, and then the ventilation area is increased from 0.741 to 0.7605. The airside pressure drop can absolutely be decreased; also, the A and K value can be increased respectively by 2.5%, 3%. That`s confirm again our sample perfectly reaches the requirements.

- Coolant Pressure Drop analysis

  In the case the tank structure is constant **$\Delta Pint=kV^2$**.

ΔPint ——Coolant Pressure Drop
k——Coefficient
V ——Flow velocity per tube

$V_1=16/37=0.43$
$k=\Delta pint_1/V_1^2=13.41$

$V_2=16/38=0.42$
$\Delta pint_2=kV_2^2=2.37$

TABLE V.HBR SIMULATION DATA

Operating point	Engine rmp	Vehicle speed	Air speed	T°amb	Q liquid	T° coolant inlet	A/C	Pth mini target	Pth HBR
	[tr/min]	[km/h]	[m/s]	[°C]	[L/h]	[°C]	[kW]	[kW]	[kW]
Point 1	2845	50	3.07	32	3646	118	0	37.7	46.6
Point 2	3003	77	3.52	37	3849	118	0	38.5	48.3
Point 3	3432	88	3.71	45	4398	115	9	43.4	43.7
Point 4	3591	110	4.17	45	4602	115	9	45.2	47.9
Point 5	3510	90	3.74	32	4498	118	0	46.3	55.7
Point 6	4244	130	4.61	37	5439	110	7.5	55.7	55.8
Point 7	5306	200	5.89	45	6800	118	0	75.3	68.8
Point 8	5306	200	5.89	32	6800	118	0	76.6	82.5
Point 9	5466	206	5.97	25	7005	118	0	78.2	91.4

TABLE VI. DATA COMPARISON

Point	Engine rm	Vehicle speed	Air speed	T°amb	Q liquid	T°coolant inl	A/C	Pth mini target	Pth HBR
	tr/min	km/h	m/	°	L/	°	k	k	k
Point 1	2845	50	3.04	32	3720.8	118	0	37.7	49.1
Point 2	3003	77	3.49	37	3927.4	118	0	38.5	50.8
Point 3	3432	88	3.67	45	4488.5	115	9	43.4	45.7
Point 4	3591	110	4.12	45	4696.4	115	9	45.2	49.7
Point 5	3510	90	3.70	32	4590.5	118	0	46.3	58.0
Point 6	4244	130	4.55	37	5550.4	110	7.5	55.7	57.2
Point 7	5306	200	5.78	45	6939.4	118	0	75.3	69.1
Point 8	5306	200	5.78	32	6939.4	118	0	76.6	82.7
Point 9	5466	206	5.86	25	7148.6	118	0	78.2	91.5

Q liquid	T liquid at inlet	Vair on core	Tamb	ΔPint max GOAL	ΔPint HBR
[L/h]	[°C]	[m/s]	[°C]	[mbar]	[mbar]
3000	95	5	20		99
5000	95	5	20		213
6000	95	5	20	300	293
8000	95	5	20		506
10000	95	5	20		796

Q liquid	**T liquid at inlet**	**V air on core**	**Tamb**	**ΔPext max GOAL**	**ΔPext HBR**
[L/h]	**[°C]**	**[m/s]**	**[°C]**	**[mbar]**	**[mbar]**
6000	**95**	**3**	**20**		**94**
6000	**95**	**5**	**20**		**196**
6000	**95**	**6**	**20**	**200**	**254**
6000	**95**	**8**	**20**		**385**
60000	**95**	**10**	**20**		**556**

TABLE VII. HBR TEST DATA

Operating point	**Engine rmp**	Vehicle speed	Air speed	T°amb	Q liquid	T°coolant inlet	A/C	**Pth mini target**	**Pth HBR**
	[tr/min]	[km/h]	[m/s]	[°C]	[L/h]	[°C]	[kW]	[kW]	[kW]
Point 1	2845	50	**3.04**	32	**3721**	118	0	37.7	49.1
Point 2	3003	77	**3.49**	37	**3927**	118	0	38.5	50.8
Point 3	3432	88	**3.67**	45	**4489**	115	9	43.4	45.7
Point 4	3591	110	**4.12**	45	**4696**	115	9	45.2	49.7
Point 5	3510	90	**3.70**	32	**4591**	118	0	46.3	58
Point 6	4244	130	**4.55**	37	**5550**	110	7.5	55.7	57.2
Point 7	5306	200	**5.78**	45	**6939**	118	0	75.3	69.1
Point 8	5306	200	**5.78**	32	**6939**	118	0	76.6	82.7
Point 9	5466	206	**5.86**	25	**7149**	118	0	78.2	91.5

Δ Pint is representable when air speed is at 5m/s, and Δ P ext

is representable when coolant flow is at 6000L/h. To evaluate our sample heat rejection, we will perform the same simulation with KULI software and analysis with the Multi-functional test bench as our standard prototype. The analysis consists to perform work test bench and simulation analysis of our radiator sample for nine different vehicle speeds (point 1 to 9). From our experience results as shown in tab 5-6-7 we can observe except the point 7 that our second prototype can meet the standard sample heat rejection requirements (ΔPint and ΔPext).

III. CONCLUSION

In this paper, we change the radiator tube design to improve the heat dissipation performance. Our analysis results show that the modifications we made on the tube have significant impact on the radiator heat rejection, and also increase his structure strength. But we still have to further our study on the radiator working performance (heat rejection, structure fatigue durability, strength ect), because the car industry have more and more requirements on the engine working, and the radiator is the most important part on the cooling system.

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