DEVELOPMENT OF AN AUTOMOBILE EXHAUST DRIVEN AIRCONDITIONING SYSTEM FOR A CAR

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Abstract The present work is an attempt made to develop, design , fabricate , experiment and analyse the possibilities of the use of exhaust of an automobile engine to run a turbo-booster which in turn drives the generator to give the required output to run a compressor and so the A/C system existing in a car.

Keywords: Exhaust, Energy, Car A.C.

INTRODUCTION

An attempt has been made to use the exhaust of the automobile (car) to run the air-conditioning system in a car. The whole system is developed and tested on an experimental basis on a Fiat car engine in the laboratory and the results obtained are quite encouraging. It offers cooling at a fuel consumption rate which is 30 % less than that obtained with conventional A/C system.

The details ,design , fabrication, experimental work and results and discussion are given in the subsequent paragraphs.

LITERATURE SURVEY

There are about three types of Automotive Air-conditioners . [S.C.Arora,S.Domkundwar,1996]

1 Expansion Valve System
2 Accumulator – Orifice Tube System
3 Suction Throttling Valve System

Expansion Valve System
This is the earliest and probably the best . It uses an expansion valve to control refrigerant flow and cycles the compressor clutch to control evaporator temperature. The main components are compressor condenser , Receiver /Filter/Drier, Expansion valve, Evaporator and Thermostatic switch. An air-conditioning system has two parts each with a different pressure.High pressure side starts at the compressor discharge then condenser and the drier and to the expansion valve.

Low pressure side starts at the output of the expansion valve ,goes through the evaporator and back to the compressor . (a well designed air-conditioning system should not have a drop in pressure due to too small lines)

Accumulator – Orifice Tube System
Uses a fixed orifice and an accumulator to control refrigerant flow and cycles the clutch to control evaporator temperature .The orifice is not as small or as large as an expansion valve could be . There is no control of too much refrigerant. The accumulator replaces direr and has one or two desiccant bags inside .{ accumulator does not work as well as filter drier at removing moisture or acid, hence two bags }. Accumulator is fitted on the suction side of the system between output of the evaporator and the input of the compressor . The accumulator has a stand pipe in it that functions by not allowing liquid refrigerant to go back to the compressor .The preset thermostatic switch or a pressure cycling switch helps slugging and controls evaporator temperature . The pressure cycling switch seems to work better than the thermostatic switch.This system is used in GM and Ford.

Suction Throttling Valve System.
This system uses an expansion valve to control refrigerant flow out of the evaporator. This system does not cycle the compressor, rather it cycles the compressor suction to the evaporator .

They all control evaporator pressure and temperature by controlling compressor suction to it.Suction throttling valve is located between the compressor and the evaporator .If the evaporator pressure drops below its setting, compressor suction is cut off to the evaporator . There is still input to the evaporator from the expansion valve so the pressure rises and the suction is re-applied. This happens so quickly that it is a throttling action that controls evaporator pressure. By controlling the minimum pressure in the evaporator one can control the minimum temperature of the evaporator. This system is used in General motors, Ford and Chrysler cars.

Concept ,Description and Laboratory Setup

The basic concept is to use exhaust gas energy to drive a turbo booster (oil cooled) which in turn drives a generator coupled with it. Electricity so produced drives the compressor of the air conditioning system.

The system utilized is EVS which is the existing system of the car. The compressor used at present in the car is replaced by an assembled positive displacement sealed compressor. To give shape to the concept a laboratory setup was prepared on fiat engine having 4 cylinders, 1100 cc with C.R as 7:1. The maximum pressure was measured at the cylinder outlet under cutoff condition which was 8 bars. Exhaust pressure at 2 feet from manifold junction was 4 bars and the temperature 250 ^{0} C.

The design of the nozzle and turbine was done on the basis of exhaust pressure ,temperature , velocity at the throat and other parameters . The throat diameter was determined based on velocity. Impeller design was done based on various parameters and so mean diameter , blade dimensions and its angle was determined based on the existing design methodology.[Cohen h.,Rogers. 1980] The casing was designed considering the impingement and smooth exit of the exhaust gases. Fabrication was carried out in the workshop with the help of part print analysis. Details of the data taken for design and final dimensions of the turbine are given in the appendix and figure 1. gives the setup prepared in the laboratory.

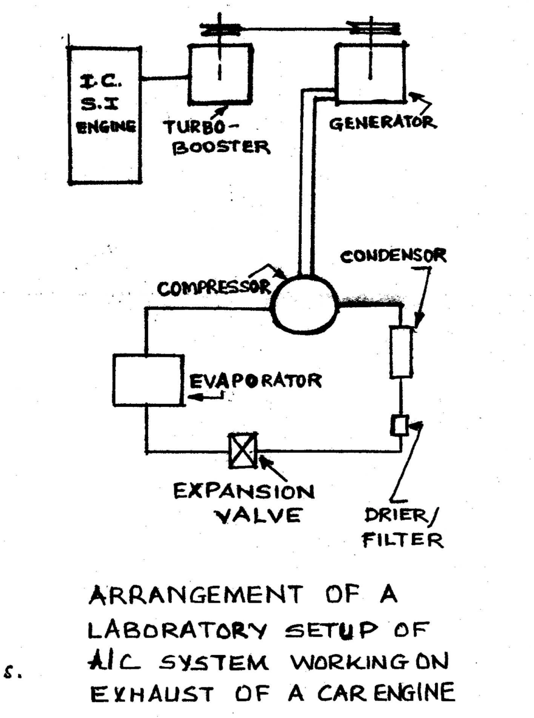

Fig. 1

EXPERIMENT AND RESULTS

The turbine was first mounted on a rail which was fitted near the exhaust outlet of the fiat engine.The nozzle fabricated as per the design was fitted in the exhaust pipe with a bypass opening. The engine was run and the exhaust was impinged on the turbine .Readings of the engine rpm ,turbo-booster rpm and generator rpm (which was driven by a belt connected to the turbine pulley) was recorded. When the generator output reached the peak the compressor connected with it started running and within a reasonable time span gave the required cooling .

Table 1. gives the results obtained under different exhaust condition for turbo-booster rpm and generator output while table 2 gives the details of cooling rate using the present system. To compare the results obtained in the laboratory a study of the cooling condition in a car was carried out.Case study carried on 800 cc car with cabinet area 3.59 meter cube.

Reduction in temperature was from 34.3^0 C to 26.4^{0}C in 10 mins., at that time atmospheric temp. outside and inside the cabinet was 35.1^{0}C and 34.3^{0}C respectively. During the measurement the car was parked in hot shade with window glasses closed and cold air re-circulated. Fuel consumption when A/c was in use increased upto 20% to 30 %. The electromagnetic clutch remains engaged during use, so battery / alternator undergoes constant energy drain.

This comparison clearly shows the worthiness of the present system, as it does not consume any additional fuel for its operation.

Table 1

Exhaust Press.	Turbine rpm	Generator rpm
6 bars	3674	2134 220 V AC
5 bars	2963	2050
4 bars	2060	1640
3.5 bars	1050	890

Table 2

Readings of A/C setup
Cabinet volume = 3.0 m^3
Exhaust pressure = 6 bars
Turbo-booster rpm = 2674
Generator rpm = 2134
Gen. Output 220 V / 2 amps.
Ambient temp. = 33 ^{0} C

Time in mins.	Cabinet Temp. ^{0} C
10	30
20	28
30	25.1
40	19.8
50	16.5

CONCLUSION

System works satisfactorily. The turbine rpm (driven by exhaust of the engine) is about 3674 rpm while the generator rpm is 2134 and gives an output of 220 V and 2 amps. This output is sufficient to drive the compressor of the EVS system.

The cooling obtained in the model cabinet prepared was 16.5 ^{0} C in a time period of 50mins. When ambient temperature was 33 ^{0} C .

Fuel consumption readings are taken for a standard car of 800 cc fitted with a conventional air conditioning system (compressor operated by electromagnetic clutch). The results have shown that fuel consumption increases by 30 % when A/C is ON under stationary condition . When the compressor is replaced by the turbo-booster /generator /compressor system designed ,the fuel consumption rate is same as when the A/C is OFF with the conventional system. This means that by using the exhaust based system 30 % of fuel saving with comfort A/C can be achieved.

Future scope

Future work can be done in redesigning of the turbine to give high torque at low rpm and pressure to meet the load conditions.To eliminate back pressure problem in the engine , a bypass valve can be provided which can maintain the impinging pressure to the turbine. Effective and efficient heat transfer methods can be used so that further cooling can be achieved.

APPENDIX

Values of various parameters obtained for turbine based on design.

Outlet pressure at throat	P_0 =1.9 bars
Outlet temp. at throat	T_0 =492 K
Throat velocity	V_0 = 425m/s
Specific volume at throat	v_0 = 0.77 m/kg
Isentropic mass flow rate	M_{isen} =0.05 kg/sec
Throat area	A = 8.08 x10^{-5} m
Throat diameter	d =10.44 mm
Temperature at the exit due to Isentropic expansion from nozzle	T_b = 419^0 K
Actual temp. due to isentropic Expansion at exit	T_b' = 457^0 K
Specific volume due to isentropic Expansion at the exit	v_b' = 1.27 m^3 /kg
Exit velocity to isentropic Expansion at exit	V_b' = 507 m/s
Actual area of the nozzle	A = 1.2 x 10^{-3} m
Actual exit diameter of nozzle	d_0 = 39 mm

Determination of mean diameter of Turbo-booster

Temp. drop. Coeff.		=	3.5
Degree of reation		=	0.3
Flow coeff.		=	0.8
Swirl angles exit	α_3	=	22^0
Rotor blade & outlet angle	β_3	=	52^0
Swirl angle at inlet	α_2	=	60^0
Axial flow velocity at Inlet	C_{a2}	=	270 m/s
Mean dia. of Turbine	d_m	=	0.2m
Blade height	h	=	0.05m
Root radius	r_r	=	0.083 m
Blade width	W	=	16mm
No. of blades		=	36
Load considered on the blade			
Tangential load	F_t	=	62.3 N
Impinging force on blade	F	=	180 N
Axial load	F_a	=	168 N

Data taken for Turbo-booster design

Max. Pressure	5 bars
Min. Pressure	3 bars
Back Pressure	1 bar
Discharge coefficient	0.99
Nozzle efficiency	0.75
γ	1.33
C_p	1.11 KJ /Kg^0 k
Isentropic efficiency	0.75
Nozzle loss	0.05
Degree of reaction	0.28

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