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THERMAL ANALYSIS & TESTING ON (AW-TYPE) HERMETICALLY SEALED RECIPROCATING REFRIGERENT COMPRESSOR

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ABSTRACT

The main aim of this project is to rectify various losses of the temperature at the of the surrounding of the compressor that the effect of the temperatures are more at the time of running of the compressor, these can rectifying at the time of running the compressor and identifying the temperature at various locations and design modifications are can be done for the new design conditions.

In this paper we discuss that understanding of fluid and heat transfer phenomena in hermetic compressor is very important for product engineering. Thermal mapping provides more realistic inputs too analytical model that simulate various compressor components. It also helps in predicting the performance changes easily, for a given change in design. The current work deals with the use of commercially available FEM codes to simulate temperature and pressure distribution in the entire compressor domain, followed by experiments to validate the simulation results. The simulations are done by using ANSYS/FLOTRAN module and the experiments are done by using wire thermocouples for temperature measurements, capillaries and pressure gauges for pressure measurements. The simulation results are obtained at various locations are upper overhang. Stator stack, suction muffler inlet, and lower overhang.

And also in these paper we done the thermal analysis of the compressor of various locations and we find the EER, and also the Electric and mechanical losses at various parts, it will benefits the product designer at the time of manufacturing.

Keywords: THERMAL MAPPING, EER

INTRODUCTION

The complex fluid flow and heat transfer phenomena in hermetic compressor are very difficult to analyze theoretically. Because there is insufficient understanding of the physics involved, assumptions are made in order to solve these problems analytically, and these assumptions can have a negative impact on the quality of the results. Hence it is necessary to carefully simulate the heat transfer inside the compressor, since it governs the energy efficiency of the whole system. The most important contributors to the energy efficiency are the suction gas superheat, which is mainly related to the motor heat losses and the heat of discharge gas. Other heat sources inside the compressor are due to rotor and frictional losses. To execute any efficiency improvement programs it necessary to have knowledge about theselosses.

Objectives of the present work

The present work deals more about the application of the results obtained by simulating a whole compressor model using commercially available CFD code, ANSYS FLOTRAN. For this analysis, whole compressor model is selected as the computational domain and meshed with FLUID 142 element. The material and boundary conditions are applied and domain is solved by using FLOTRAN solver to satisfy continuity, momentum and energy conservation equation. This powerful tool along with faster and robust digital computers makes it possible to predict temperature distribution in any plane across the whole domain.

The results of the numerical simulation are validated by using an experimental setup for thermal mapping of the compressor with conventional thermocouples to measure temperature and capillaries to measure pressure. Temperature and pressure at critical points are measured in the experiment. These experiments results are used to validate the simulated model. Since the simulations are matched with the experiment, any intended modification in the physical model can the incorporated in the simulation and predict the performance. This will help in reducing lot of time in engineering and in improving the quality of new design.

Thermal auditing:

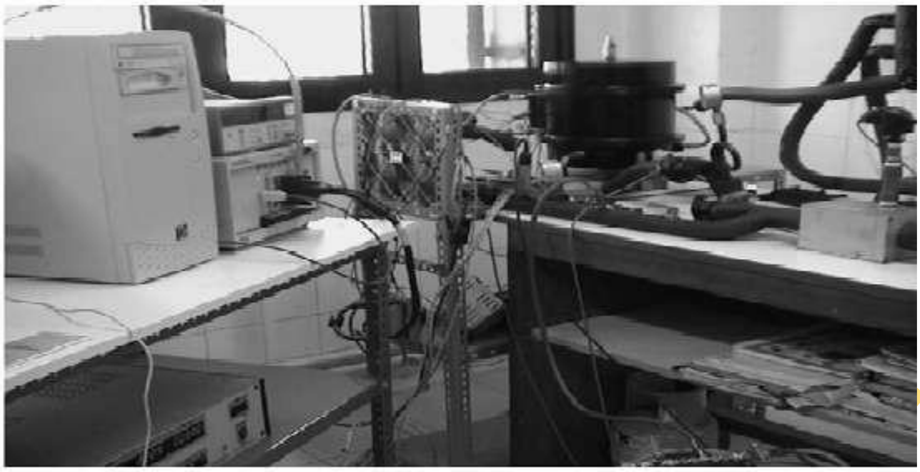

Fig:1: thermal auditing is done in thelaboratory

ENERGY BALANCE	BTU/lbm	Watts	Percent
Heat absorbed by suction gas	8.0938	596.64	31.47
“Isentropic” Work done in the cylinder	17.9242	1321.28	69.69
“Isentropic” Cylinder Heat Transfer	-0.4879	-35.97	-1.90
Minimum Work done on Gas in Cylinder	17.8391	1315.01	69.36
Maximum Cylinder Heat Transfer	-0.4871	-35.91	1.89
Maximum Work done on Gas in Cylinder	17.9234	1321.23	69.68
Minimum Cylinder Heat Transfer	-0.4028	-29.69	-1.57
Enthalpy increased in cylinder	17.4363	1285.32	67.79
Heat absorbed by Discharged Gas	-6.5610	-483.64	-25.51
Enthalpy increase across Compressor	18.9691	1398.31	73.75
Heat convected from Housing	6.7515	497.69	26.25
Energy input to compressor (Electrical)	25.7206	1896.00	100.00
Heat liberated from Electro-Mechanical losses based on “average” Work done in Cylinder	7.8394	577.88	30.48

Temperature of Isentropic Compression: Cavity 268.321 deg.F
IsentropicEfficiency: Cavity-Cavity.................................. 69.688percent

Temperature of Isentropic Compression: Inlet 216.467 deg.F
IsentropicEfficiency:Inlet-inlet.. 60.029percent

Compression Efficiency based onEnthalpyIncrease............... 67.791percent
Fraction of electro-mechanical losses
Convected fromthehousing 0.86123
Combinedelectro-mechanical efficiency 69.521percent
Calculated Electrical Efficiency
based On EstimatedMechanicalEfficiency................. 77.246percent

EXPERIMENTATION

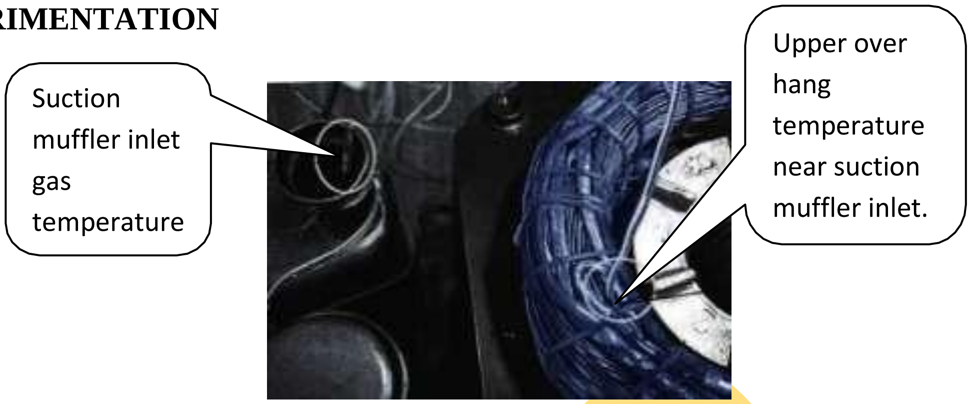

Fig:2: Upper overhang temperature near suction muffler and SMA inlet temperature

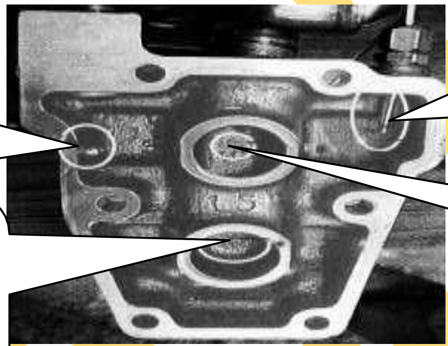

Fig 3:Plenum temperatures and pressures

Fig 4: Compressor top view with thermocouples

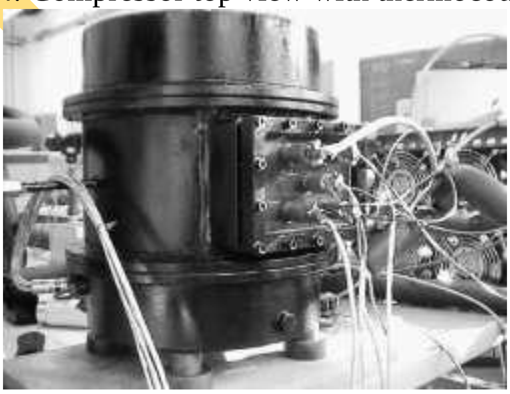

Fig5: final view of compressor with thermocouples and capillary devices

Compressor testing, Connecting the compressor, Evacuation and charging Refrigerent

The compressor after fixing the thermocouples and pressure gauges has to kept it in the compressor chamber of the calorimeter. The suction and discharge tube connections are to be made, depending upon the capacity of the compressor. Wells are provided for inserting return gas and discharge temperature RTD sensors. The flexible hoses with threaded adaptors are provided for connecting the compressor quickly in the chamber to the refrigeration system. After connecting the compressor to the primary refrigeration system, follow the procedure given below

1. Ensure that all the valves provided in the primary refrigeration system are fully open. The valves are fully open. The valves are provided at the followinglocations.
   a. Compressor Discharge and Compressor Suction.
   b. Valve in the liquid line after the Mass FlowSensor.
   c. Valve at the receiver outlet.
   d. Valve at the drier outlet.
2. Pressurize the system with dry nitrogen to a pressure of 25 kg/cm^2. Check the joints you make for leakage. While applying the nitrogen pressure, close the suction Gauge by operating the hand shutoff valve provided below the gauge to prevent the damage to the gauge on account of the very high nitrogen pressure. You may pressurize the system up to 20 kg/cm^2 pressure and shut off the valve before increasing the nitrogen pressure further.

The pressure transmitter for suction pressure has the range of 0 to 17.23 bar. Maximum pressure that can be applied to the sensor is 4 times its full scale.

3. Release the nitrogen pressure from the primary system and connect a good quality two-stage, rotary, high vacuum pump at both, the suction and the discharge process connections provided in the compressor chamber. Evacuate the entire system to at least 75microns.

A good quality deep vacuum gauge is a must during the evacuation of the entire primary system or only compressor. It is more important that how deep the vacuum was instead of how long the system was evacuated.

Evacuating the system from both the suction and the discharge side is essential since the expansion valve provided on the primary system may remains fully closed at standstill depending upon the shutdown condition.

4. Break the vacuum with desired refrigerant. There are three sight glasses in the liquid line. The first one is at the receiver, the second is just before the mass low sensor and the third is before the expansion valve, that is, inside the secondary pot chamber. During charging, ensure that the sight glasses before the mass flow sensor and the expansion valve are completely filled and the receiver sight glass is at the halflevel.

It is not necessary to release all the gas from the primary system every time. The compressor can be isolated from the rest of the system by closing the suction and discharge valves. Only compressor needs to be evacuated after changing the same.

Testing the compressor

After charging the refrigerant into the compressor the calorimeter is connected to the inverter to get the different load conditions. The thermocouples are connected to digital unit and pressure probes are directly connected to the pressure gauges. Compressor started and the experiment is conducted in the different load conditions. The readings are taken after attaining the steady state in every load step. All the readings at various loads tabulated in the table 8.1

Another experiment is conducted to measure the shell temperatures at various locations on the compressor shell. A production AW compressor is taken and marked the horizontal lines and vertical lines using marker on the outer side of the shell. At the intersections of horizontal lines and vertical lines the temperatures are measured. As described earlier compressor is fitted to the calorimeter and charged. This experiment also conducted at various loads. All the temperatures readings over the shell are taken by optical pyrometer at the intersections of the horizontal and vertical lines. All these readings are tabulated in the table 8.2.

THE RESULTS OBTAINED FROM THE CALORIMETER TESTING AT VARIOUS LOADS:

		185 V	185 V	195 V	220V	230 V	230 V	
Location	Thermocouple number	35Hz	40Hz	50Hz	60Hz	65Hz	70Hz	10 min after

								switch off
Suction Muffler inlet	2	53	49	47	47	47	48	63
Suction plenum	12	94	91	91	91	91	91	69
Upper overhang	3	41	38	36	35	35	36	65
Stator stack	11	17	66	66	64	65	65	67
Lower overhang	5	77	70	70	69	69	70	65
Resonator out	7	105	103	105	106	107	107	66
Discharge out	1	105	102	101	102	103	104	65
Upper overhang @ SMA	4	48	44	41	42	42	42	65
Oil temperature	8	86	84	87	86	86	85	66
DMA out	6	117	115	114	115	115	117	63
Discharge plenum	Rod thermocouple	125.4	118.4	121	121	121.1	126	67.4
Liquid temperature		46.1	45.6	45.5	45.8	45.6	45.7	
Discharge temperature		108.5	110.5	112	114.4	115.8	117.5	
Top shell temperature		43.5	41.2	41.6	41.5	41.7	41.7	
bottom shell temperature		56.1	55.9	59	61.4	59.3	61.7	
Ambient temperature		35.2	35	35.2	35	35.1	35	
Return gas temperature		35..1	35	35	35	35.2	35	
Pressure gauges								
Dial gauge 1	Discharge plenum	312	316	318	322	323	326	154
Dial gauge 2	DMA exit	310	312	314	318	317	319	153
Pressure sensor	Suction plenum	72	72	72	72	71	71	155
Suction pressure		75.5	75.5	75.5	75.6	75.6	75.6	
Discharge pressure		300	300	300	300	300	300	
Plot pressure		125	125	130	140	150	140	
Mass flow rate(Lb/hr)			150	200	240	260		
Total power(watts)		1420	1522	1912	2220	2427	2568	

Table 1: Conducted on DAW 5524 compressor

Pressure sensors used for measuring suction plenum and discharge plenum pressure are least count 1 PSI mass flow rates are measured for reference only. May not be accurate enough because of flow metre accuracy.

Temperatures over the shell measured at various locations on the shell in calorimeter using pyrometer.

Location	40Hz	50Hz	60Hz	65Hz	70Hz
1	56	50	50.8	49	52.2

4	52	43	52.1	48	55.1
5	43.9	37	42	41.6	44.2
6	56.5		54.2	51.8	55.9
9	53	45	49.9	50	51
10	54.8	45	56	50.4	56.1
11	64.9	56	65.5	63.5	68.5
14	60	50	59.7	58.1	62.3
15	66.6	55	68.1	69.5	72.4
16	82.5	74	85.3	85.1	88
19	83.5	73.5	83.7	83.4	89.4
20	83	75	86.9	88.3	89.4
21	81.5	76	85.6	82.7	82.2
24	81.5	76	85.1	85.6	86.4
25	72	86	86	87.5	90.1
A1	49.8	48.5	49.5	49.4	49.3
A2	49.8	48	48.4	48.4	45.4
A3	53	53.5	55	53.3	57
A4	72	73.5	71	66.5	78.2

Table 2:Temperature at various locations on the shell

Simulation results:

The proposed numerical simulation is carried out for hermetically sealed reciprocating compressor used for Air-conditioning applications. The geometry of the entire compressor is modelled using PRO-E. Because of the geometrical complexity of the domain, the model is simplified by removing unnecessary fillets, restrictions, sharp edges and sharp corners. So that it is easy to mesh the domain in pre-processing stage. Mainly the domain of interest in this simulation is temperature distribution over motor and pump assembly. The domain consists of compressor shell, stator stack along with copper winding, crank case and cylinder head. The suction tube is also modelled in order to allow the refrigerant gas to enter into the compressor shell.

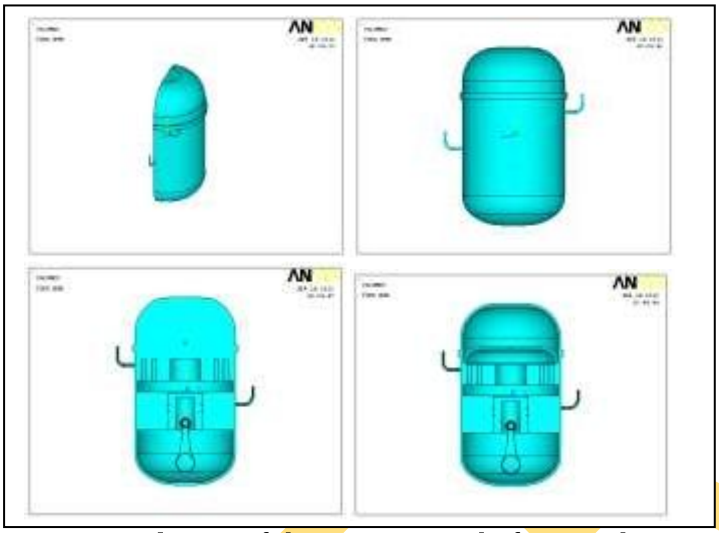

Fig: Volumes of the compressor before meshing

The ANSYS FLOTRAN contains two types of elements, FLUID141 and FLUID142 . FLUID 141 is a 2-D element and FLUID142 is a 3-D element. This two elements solve for 2-D and 3-D flow, pressure, and temperature distributions in a single-phase viscous fluid. For these elements, the ANSYS program calculates velocity components, pressure, and temperature from the conservation of three properties: mass, momentum, and energy.

For the present simulation

1. Mass flowrateisobtained fromthecalorimeter testreportsofAW compressors.Fromthatvelocityis calculated.

2. The copper and iron losses are obtained from the dynamometer testing and from thecompressor programs likeHPUMP.

3. Oil temperatures are taken from calorimeter testreports.

4. The parameters like external heat transfer coefficient is calculated using standardcorrelations.

The calculation of velocity from the given mass flow rate:

Mass flow rate= 1.871 Kg/min.

= 0.0311833 Kg/sec.

Area of inlet = 1.2828E-0.4 m^2.

Density of the R-22 gas = 19.05Kg/m^3.

Mass flow rate = density(ρ)*area of the inlet(A)*velocity(V)

Velocity = 12.76 m/sec.

The copper and iron losses = 323 w.

Oil temperature = 80^{o}c

External heat transfer coefficient = 5 W/m^2-K.

Fig 7.4: Boundary conditions applied on thecompressor

The velocity in the directions V_x, V_y, V_z and suction gas temperature T are defined at the inlet. The heat sources are defined as volumetric heat sources on the motor & pump. The out flow boundary conditions is used at the outlet, which is a relative pressure (usually zero). In the absence of the outlet, which is a relative reference frame, the absolute pressure is the sum of the FLOTRAN (relative) and the reference pressure. The volumetric heat source is applied on motor & pump. Oil temperature is applied on bottomsurfaces.

Problem setup and solution

In the sequential solution algorithm, there are three options for solving equations sets for degrees of freedom:

1. A fast, approximatesolver

This solver, the Tri-Diagonal Matrix Algorithm (TDMA), Performs a user-specified number of iterations through the problems domain.

2. “Exact”solvers

The “exact” methods are semi-direct conjugate directions methods that iterate to a specified convergence criterion.

These methodsare:

- The conjugate Residual (CR), preconditioned conjugate Residual (PCCR), preconditionedGeneralized Minimum Residual (PGMR), Preconditioned BiCGStab (PBCGM) methods for non-symmetric matrix equations.
- The preconditioned conjugate gradient method for the incompressible pressureequation.

3. Sparce Directsolver

This solver uses Gaussian elimination to factorize the matrix and then uses backwark/forward substitution to solve for the unknowns.

The default solver (TDMA) for the velocities and the turbulence equations is adequate for virtually every problem encountered. In general, the solution of the pressure equation must be accurate and conjugate direction methods are used. However, the TDMA method can be successful for natural convection flows. After defining the solvers, the environment variables have to define. They are reference conditions and gravity in order to solve the governing equations. The reference conditions are reference pressure, bulk modulus parameters ratio of C_p/C_v, nominal temperature, stagnation temperature and bulk temperature. And also stability parameters are to be defined, to solve the momentum, continuity energy equations.

Choosing the appropriate turbulence model is an important step, which depends on the nature of the gas flow inside the suction and discharge paths, i.e. highly turbulent flow, rotating or flow with strong coupling between the turbulent quantities and the mean flow quantities. The standard K- ε model is used in present simulation, which is robust, economic, and gives reasonable accuracy for a wide range of turbulent flows. The turbulence parameters such as turbulence intensity and hydraulic diameter are specified. The domain is initialized basedon inlet boundary and the simulation is carried out in steady state condition for sufficient number of iterations to get theconvergence.

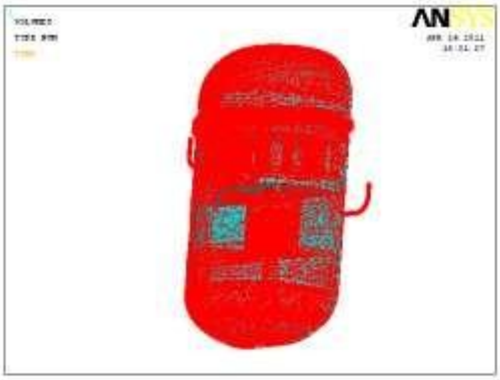

Fig 7.5: Temperature distribution over the compressor shell

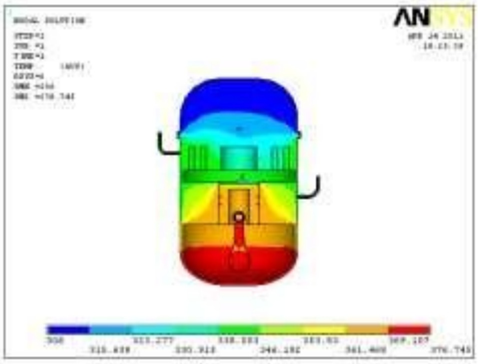

Fig 7.6 : Sectional View of the compressor alone the working plane

SimulationResults

The results are obtained after solving the problem up to convergence limits. The results taken are temperature distribution across the compressor and velocity vector plots. Following are the results obtainedfrom the ANSYSsimulation.

Fig 7.5 shows the temperature distribution over the compressor shell. Fig 7.6 and 7.7 shows the front and back views of the compressor. From this figures, the temperature distribution over the shell is observed from the range of 308^{o}K to 376.745^{o}K . the lowest temperature at the top shell and higher temperature are at the pump region

Fig 7.6 Shows the details of the temperature distribution at high temperature regions at the top end that is ranging from 369 ^{o}K to 376.745 ^{o}K. But most of the are is at temperature around 375 ^{o}K . in these fig 7.7 and fig 7.6 shows the temperature distribution over winding, stator, and lower winding are observed and the sectional views are taken normal to screen to observe the temperature distribution over the variousregions.

CONCLUSION

This project demonstrates the strength of computational fluid dynamics in simulating flow and thermal characteristics of the overall compressor in steady state conditions. A numerical model for an Air conditioning compressor is developed and analyzed using commercially finite element package ANSYS. The entire computational domain is solved for conjugate heat transfer and the results are seen across different planes. It is seen from the results that the numerical results are in good agreement with the experimental results.

Besides CFD is used here at the design stage of the compressor, there by reducing the number of prototypes for trail and error and hence the reduction of total design cycle time. The visualization tools like velocity plots;

temperature and pressure contours are used to streamline the gas flow by modifying and improving flow passages in plenums, mufflers and shockloop.

Comparison between experimental results and simulation results.

Measurement location for temperature (^{0}k)	Experimental Result	Simulation Result
Temperature on upper overhang	308	313.941
Upper overhang@SMA	315	314
Stator Stack	334.5	338.353
Suction Muffler Inlet	320	316
Lower overhang	340	338.6

Table 9.1 : Comparison between experimental and simulation results

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