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Improving Gas Turbine – HRSG output using Inlet Air Chilling and Converted Evaporator

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Abstract - *Every machine ever created has some output limitations with respect to the inputs given to them. In order to change the output rate some design and condition modifications have to be done in order to get effectiveness beyond the designed parameters and at the same time also not hamper the metallurgical and operational stability of the machine. In order to get more out of a GT-HRSG, since they are temperature controlled machines, some design modifications proved a considerable increase in output keeping the efficiency at bay.*

1. INTRODUCTION

With a continuous improvement and expansion in process plants going on, there is a vital need of utilities to have more efficiency to put in more effort to give the intend output and keep production at par. Gas turbines work on the principle of hot gas expanding. They have a fixed loading capacity. In order to increase the loading, the ambient temperature plays an important role as it eventually determines the output temperature. In case of peak load, if the exhaust temperature reaches the critical value, the machine will not allow any more power loading and the machine will go into temperature control.

Another problem related to over temperature is related to HRSG. In order to increase the steam loading of HRSG there is a provision of supplementary firing. In case of elevated gas turbine exhausted temperature, added with supplementary firing the metal temperature of superheater-II tends to increase as the supplementary firing is followed by the superheater-II. The idea of improving the conditions of these machines at peak load was to improve the reference temperatures of turbine and at the same time improve the metallurgy in order to keep the HRSG healthy.

The modifications thus done was done in a GE Frame 6B Gas Turbine with a power capacity of 34.5MW. Along with this another design change was done in the 120 MT HRSG which followed.

2. INLET AIR CHILLING

The Frame 6B has a capacity of going up to a loading capacity of around 34 MW which was observed on cold winter days where the ambient temperature was less and the reference exhaust temperature (TTXM) allowed the machine to go as much as 34-34.5 MW on preselect mode, before it can go to temperature control. On summer days this can come down to as low as 29 MW and the machine is almost on the verge of accepting no more MW input.

The turbines used in this case had very high demanding complex loads and importing power was proving very expensive. To avoid this the idea of Inlet Air Chilling (IAC) was tried to see if this can prove out to be beneficial. The idea was to cool the incoming air so as to get more window for MW input and run the machine on maximum loads.

To put the idea in use, a new set of filter module was installed. It had air inlet from three sides. The fourth side was attached to a bellmouth which then ended in the inlet plenum. The filter module had chilled water tubes running throughout. The filters were placed in an order so has to have the best possible air in inlet. It had one mesh to avoid any big particle enter. Then it had a primary filter followed by chilled water tubes, and then a denser secondary filter.
A 1000 TR vacuum absorption cycle based chiller was commissioned and it could suffice for any climate conditions.

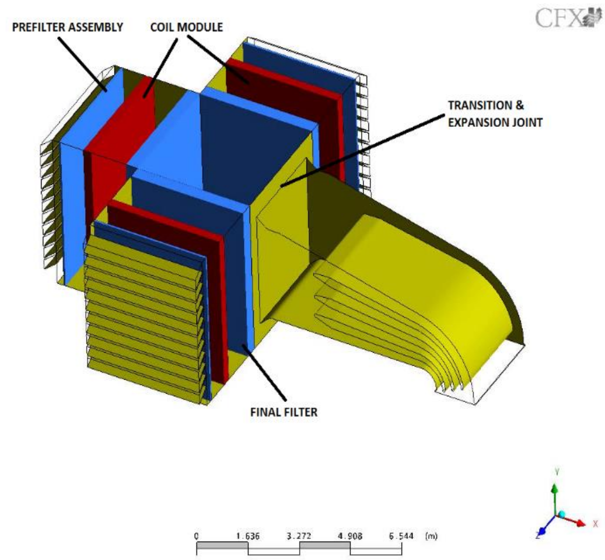

Fig-1: Solid model of Inlet air bellmouth

This filter assembly not only provided an option to have inlet air chilling but also provide uniform air flow throughout the three filter modules. The CFD analysis iterations provided a final design to achieve optimum airflow with minimum backpressure.

The following figures clearly shows uniform entry of streamlines to the aero-acoustic diffusers at 3D modeling and top view of CFD results respectively, which reduce the vortex and turbulence with in the Gas Turbine inlet duct. This would also reduce flow induced vibration in the duct.

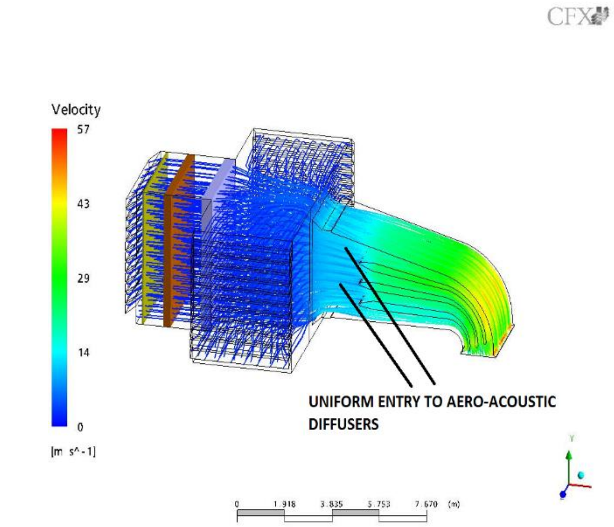

Fig-2: Velocity drop

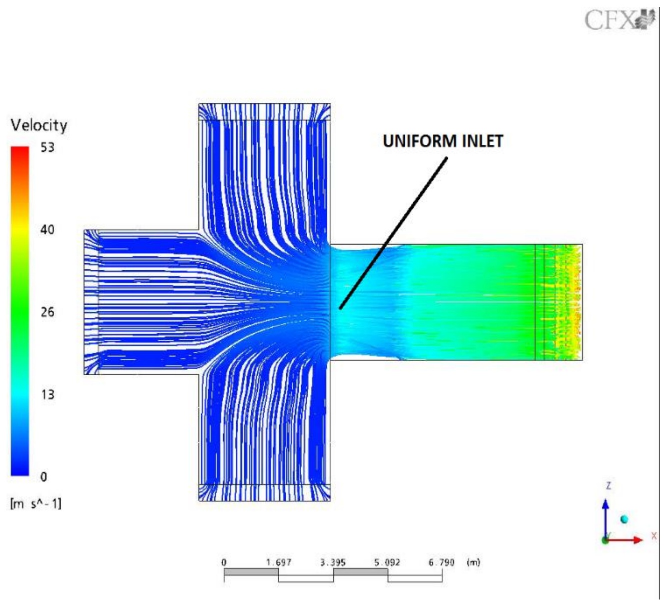

Fig-3: Velocity distribution

The analysis showed total pressure drop values, which were quite low compared to previous model. In the proposed model the pressure drop was expected to be around 1.97" WC From rain hood to the Gas Turbine inlet.

There also was significant reduction in the duct flow induced vibration due to effective diffusion of stratified high velocity air entering compressor inlet plenum. It was confirmed by having a condition monitoring team check the vibrations.

Flow Rate	130 Kg/S @ 20°C
Pre filter	0.3"WC
Water coil	0.50"WC
Final filter	0.35"WC
Pressure drop:	
Ambient – Filters	1.22"WC
Filters – Inlet plenum	0.75"WC
Total pressure drop	1.97"WC

2.1 Importance of compressor bellmouth differential pressure.

The static pressure difference between the inside of the bellmouth and static ambient pressure will give an indication of mass flow as total pressure will remain the same, the static drop converts into dynamic component (Velocity) along with flow area will give mass flow of less dense inlet air.

Mass flow = Speed * Density * Area

Pitot tube is significant in determining the inlet air flow which was vital to see the change in flow characteristics after inlet air chilling. When the Pitot tube is connected to a manometer, the resultant displacement of fluid is measure of the "Velocity Pressure".

Velocity Pressure= $P1-P2 = ½*\rho*v^2$,

ρ=density of air

$V = \text{sqrt } 2(P1-P2) / \rho$

But $P1-P2 = \rho_w * g * H$,

$\rho_{w =}$ Density of water in manometer is 1000kg/m3

H= Height of water column in meters

$g=9.81\ m/s^2$

$V = \text{sqrt } 2* \rho_w * H * g / \rho$

$= \text{sqrt } 2* 1000 * H * 9.81 / \rho$

H =height of liquid column to be expressed in mm of water h_v

$H = h_v / 1000$

$V = \text{sqrt } 2* 1000 *h_v /1000*9.81/\rho$ m/sec

So, Velocity air flow = 4.43 sqrt h_v/ρ m/sec

In practice the density of air is affected by temperature and by static pressure and so it is necessary to know the air density at some reference temperature and pressure, and then by applying corrections the value of test conditions are obtained. The density of air at 1.013 bar 0deg is 1.29 kg/m3 and it varies inversely to the absolute temperature and directly to absolute pressure.

The density ρ for any temperature and pressure condition is thus given by

$\rho = 1.29*[273/T]*(10363+hs/10363)*B/760$

Where, 1.29 is density of air at 0degC and at 10.13 bar in kg/m^3

T is air temperature, K

hs is static pressure in duct in mmWC

B is barometric pressure in mmHg

10363 is standard pressure in mmWC

So using this the condition monitoring was done with and without chilling and mass flow was thus calculated.

Gas turbine inlet mass flow without chilling:

Temperature of air inside the duct = 34 degC

Static pressure of air as per Pitot tube measurement = 18.2 (-) mmWC

Barometric pressure = 760 mmHg

Velocity at Elbow exit (from velocity distribution as per CFD) = 22.5 m/Sec

Cross sectional area of duct at Elbow = 4.61 m^2

Volume flow = 103.7 m^3/sec

Air density = 1.1451 kg/m3 at duct conditions

Thus, Mass flow = 118.74 Kg/Sec

Gas turbine inlet mass flow with chilling:

Temperature of air inside the duct = 21 degC

Static pressure of air as per Pitot tube measurement = 32.92 (-) mmWC

Barometric pressure = 760 mmHg

Velocity at Elbow exit (from velocity distribution as per CFD) = 22.5 m/Sec

Cross sectional area of duct at Elbow = 4.61 m^2

Volume flow = 103.7 m^3/sec

Air density = 1.1941 kg/m3 at duct conditions

Thus, Mass flow = 123.82 Kg/Sec

The above calculations were done against the base load design mass flow of 135 Kg/Sec. The calculations clearly indicate a higher density of air flowing after using the inlet air chilling. In case of any sweating, a condensate collection pan is provided leaving air side of each dehumidifying pan. It also had a condensate trap of at least 2.5 times the expected negative static pressure of the unit.

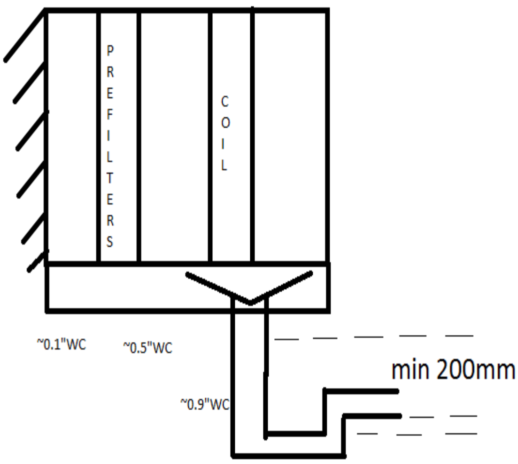

Fig-4: Coil sweating recovery U-tube

Expected negative pressure is

0.1+.05+.09=1.5" WC

d= 1.5" WC * 2.5 = 3.75"

Trap used was of 4" (100mm).

The following graph shows how the gas turbine behaves without any inlet air chilling in normal conditions.

Table-1: Gas turbine data without chilling

GT DATA WITHOUT INLET AIR CHILLING					
		Time			
Parameter	Unit	1200	1300	1400	1600
Speed	RPM	5020	5062	5057	5030
Load	MW	26.5	26.8	26.2	25.5
GT speed	%	98.55	99.37	99.29	98.75
Flow divider speed	%	68.42	68.59	67.7	66.2
FSR	%	69.16	69.04	68.38	67.54
Liq fuel flow	kg/sec	2.6	2.67	2.65	2.59
Mass fuel flow	TPH	8.13	8.19	8.03	8.02
Compressor inlet temp.	degC	30	32	34	33
Inlet air filter D.P	INWC	1.37	1.39	1.97	1.3
Compressor outlet pressure	Bar	8.7	8.79	8.67	8.52
I.G.V. Angle	DGA	83.9	83.9	83.9	83.8
Exhaust Temperature	degC	573	574	575	576
Spread Limit	degC	69.4	69.4	69.4	69.4
HRSG flue gas inlet pressure	MMWC	274	276.7	273	262
Compressor discharge temperature	degC	366	371	373	372
Heat rate (open cycle)			3169	3169	3169
Steam/fuel ratio			19.67	20.01	18.23

The following graph shows how the gas turbine behaves with inlet air chilling in normal conditions.

Table-2: Gas turbine data with chilling

GT DATA WITH INLET AIR CHILLING					
		Time			
Parameter	Unit	1200	1300	1400	1600
Speed	RPM	5018	5047	5030	5010
Load	MW	26	26	26	26
GT speed	%	98.5	99.09	98.75	98.35
Flow divider speed	%	65.12	65.53	65.2	65.53
FSR	%	66.45	65.79	66.6	67.11
Liq fuel flow	kg/sec	2.56	2.56	2.57	2.57
Mass fuel flow	TPH	7.93	7.89	7.9	7.91
Compressor inlet temp.	degC	19	20	19	19
Inlet air filter D.P	INWC	1.52	1.54	1.51	1.47
Compressor outlet pressure	Bar	8.97	9.04	9	8.93
I.G.V. Angle	DGA	83.7	83.7	83.7	83.7
Exhaust Temperature	degC	526	522	524	526
Spread Limit	degC	172.7	176.6	176.7	177
HRSG flue gas inlet pressure	MMWC	297	298.6	299.6	296
Compressor discharge tempe	degC	351	353	351	350
Heat rate (open cycle)			3169	3169	3169
Steam/fuel ratio			17.7	19.18	18.22

Based on the observations taken at different time of the day, it was observed that the inlet air chilling does a considerable amount of change in the machine. Initially it reduces the FSR, which in turns also improves the fuel going inside the GT. According to the observations both noted with the flow meter and mass flow with *micro motion*, it was observed that less fuel goes inside the machine thus reducing the flow divider speeds. With low inlet temperature the compressor outlet pressure is high with low exhaust temperatures, low compressor discharge temperatures and high HRSG flue gas pressures. Thus allowing more temperature window to increase the capacity of the machine without going into temperature control.

The only downside observed with IAC is the increase in suction air filter D.P. which has to be monitored during operations. Also at the end of HRSG NOx and Sox have to be monitored to avoid self-condensing that might result in damaging of HRSG tubes.

On longer runs and more report gathering it was observed that the heat rate of the turbine also improved. Also additionally the steam generation of HRSG improved. Also at the end of HRSG the temperature was high so an additional LP evaporator was installed which could give a steam output of 6barg which was used to run the 1000 TR chiller so the entire energy was conserved quite efficiently.

Table-3: Data comparison

With Chilling						
Day	1					
Time	Fuel Totalizer	Power totalizer	Total power generater	Fuel consume	Open cycle	MW
12:15	131987	402703	80	26	3013	29
15:00	132013	402783	120	39	3013	30
19:00	132052	402903	98	32	3027	28
22:00	132084	403001	46	14	2821	31
0:00	132098	403047	197	58	3004	30
6:00	132156	403226	523	169	2995	30
Day	2					
	132372	403878	227	74	3022	30
Day	3					
	132467	404171	293	95	3006	29
Without chilling						
Day	4		685.6	223.8	3026	29
Day	5		682.6	218.9	2973	28
Day	6		670.5	220.8	3053	28
Total			2038.7	663.5	3017	29

3. CONVERTED EVAPORATOR

With higher gas turbine loads and more output required out of HRSG, supplementary firing is required to compensate for minor heat loss and add up to the heat input to further increase the steam flow and also compensated for any process abnormalities to keep the header pressure constant. With all the modifications thus made, there was a constant problem of tube failures in HRSG at certain point of time. It was observed that the super heater-II which was most exposed to the supplementary heat, was exposed to the maximum damage and failed eventually. To avoid this some modification was done in the evaporator.

The HRSG used in line with the mentioned gas turbine had a design capacity of 100 TPH capacity of steam generation at 100 barg 495°C. It consist of two economizers, an evaporator module and three super heater module. The entire HRSG ran on gas turbines outlet flue gas. To compensate or any minor heat loss or process steam requirement change, it had six burners in between super heater II and III. At peak plant loads all the burners were used with full GT load to give maximum steam output. The problem faced had a lot of effect on machine life and also demanded big shutdowns creating a problem for process requirements.

To avoid this problem, the initial two modules of superheater II were modified. They were cut and replaced with the same tubes as used in evaporator. It also had coper fins attached to it for maximum heat transfer and absorption in case of supplementary firing and the tubes ran water unlike other superheater tubes which ran steam. This created a jacket for the superheater II section and absorbed the conductive heat coming from the burner flame so as to not damage the superheater tubes.

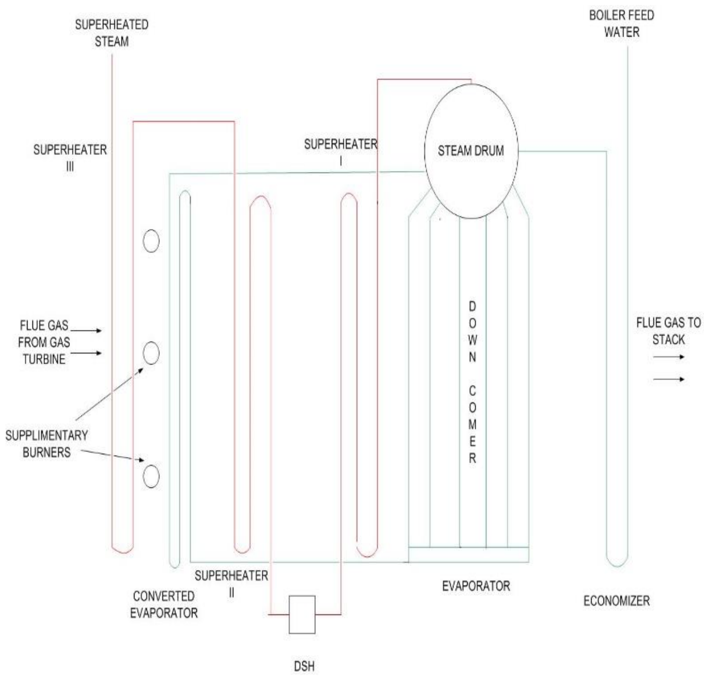

Fig-5: Converted Evaporator in a Heat recovery steam generator

Before the change, at peak loads the temperature of superheater II used to go over 515 degC. After the project, the temperature remained below 478 degC. Another benefit was that the flow output of HRSG improved ant increased to 125 TPH from 105 TPH. Additionally there was quite a bit of hunting seen in HRSG when put on flow control which also improved when kept in three element control.

The gas turbine used for the particular HRSG is a black start machine and so it's total time to reach FSNL is about 11 minutes. So the heating of HRSG can be done parallel to the startup or after it reached certain temperature. Both the cases referred to as cold start and hot start. The new tube bundle integrity was to be checked with both startups so as to confirm if it can withstand thermal shocks in case of hot startups.

It was observed that a temperature difference persisted between old evaporator and the converted evaporator. But it did not affect the metal temperature of the main superheater II tubes and served the purpose of jacketing the tube module from flame heat.

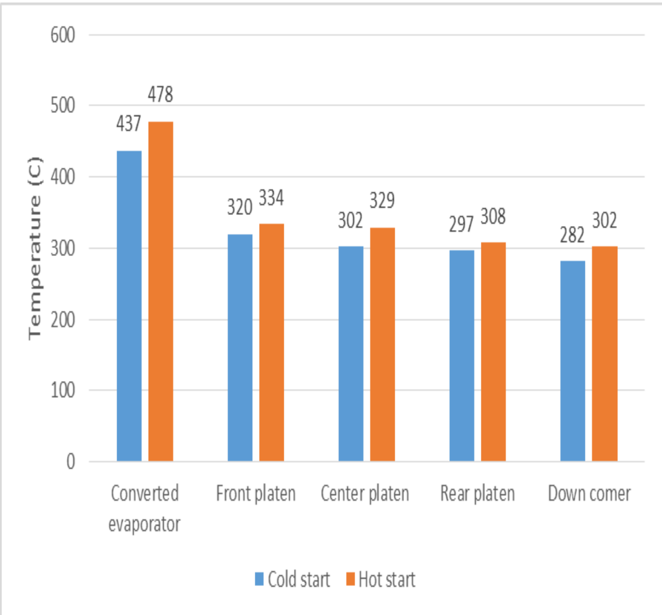

Chart-6: Fatigue effect on modification

The study showed that the maximum fatigue effect transpired in the piping at the bottom header of the evaporator, in specific, the feeders that linked the manifold to the lower headers. It was also observed that the tube bundle metal temperature was on the higher side in the transient condition during critical time period of hot startup phase compared to cold startup and operation.

The fatigue analysis results approve that the system is adequate to support the thermal growth without yielding excessive fatigue stresses.

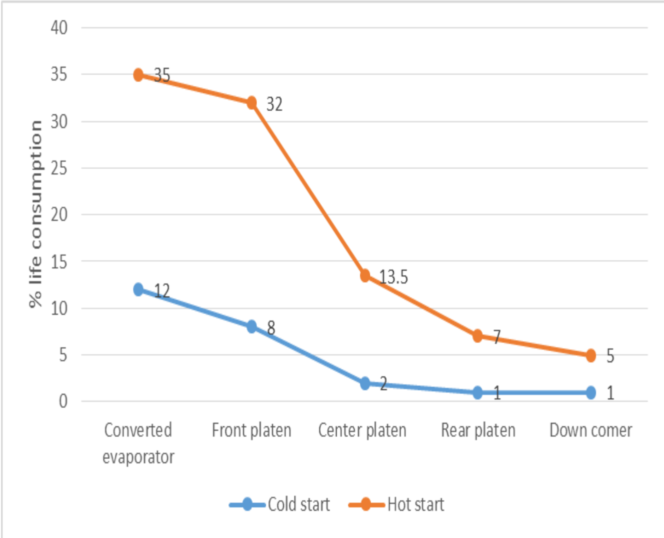

Chart-7: Effect on tube life

As the graph suggests there is a higher life consumption rate in the life of tubes, especially that of the converted evaporator as the temperature in that zone is highest for water tubes of 135 barg pressure and so it was to be avoided to take the HRSG in line with hot start procedure and was to be done only in extreme cases as required.

4. CONCLUSION

The entire project was to develop an increase in productivity in case of a gas turbine – HRSG closed cycle operation and not only did it help to increase the throughput of the machines without affecting the reliability but also made failure intensity less.

Table-4: Benefits after modification

Gas turbine IAC benefit			
Description	UOM	Values	
		Present	With IAC
GT output at ISO	MW	37.52	37.52
O/C heat rate at ISO	kcal/kwh	2764	2764
Ambient temperature	degC	35	20
Frequency	Hz	50	50
Inlet pressure drop	mmWC	100	100
Outlet pressure drop	mmWC	225	225
Correction factor for degradation/aging	%	5	5
Correction Factors:			
Temperature CF		1.0350	1.0080
Frequency CF		1.0050	1.0050
Inlet pressure drop CF		1.0000	1.0000
Outlet pressure drop CF		1.0063	1.0063
Part load CF		1.0470	1.0470
Degradation CF		1.0500	1.0500
GT heat rate for conditions	kcal/kwh	3166	3083

The overall heat rate reduction was of about 83 kcal/kwh. It also improved the loading capacity of GT in hot conditions and avoided the machine to go into temperature control.

The bellmouth design proved a huge reduction in pressure drops and uniform velocity throughout the filters.

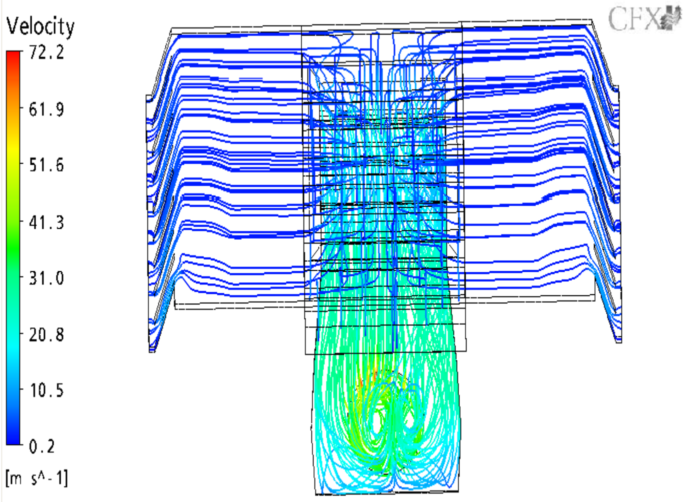

Fig-8: Pressure loss

Inlet plenum loss was calculated at two points. One in the filter house and another before the compressor inlet. At first point, a loss of 85-95 mmWC was observed and at point two, a loss of 210 mmWC was observed. According to the gas turbine vendor, a guaranteed inlet plenum loss of 101.6 mmWC was constant.

As per the performance curves, output loss for every 4 inch of H2O is 1.55% and results in a heat rate elevation of 0.5%.According to it, the GTG had an output loss of above 1.65% leading to a heat rate increase of 0.53%.

Post changing of bellmouth, a reduction of 4.9 inch of H2O in pressure drop was observed along with a power gain of 2.08 %. This led to an increase of 8000 operation hours per year and a power gain of 4664.8 MWhr. The payback time was a little under 5 months.

With a reduction in heat rate at 31kcal/kwh, the machine saw a fuel savings of 38kg/hr. with an additional steam generation benefit of 1.5 TPH.

Additional benefits from uniform flow lead to uniform loading on GT compressor blades and a significant reduction on flow induced vibrations. Also a uniform and laminar flow distribution resulted in better heat transfer coefficient resulting in better supplementary firing efficiency.

The converted evaporator completely eradicated the need of frequent shutdowns required for HRSG at peak loads and also increased the output rate of HRSG. Also the combination of tube diameter, thickness of the wall of HRSG and the thermal growth differences was adequate for hot startups. The fatigue damage will increase in the tubes due to sudden thermal shock but this will not have much effect on the planned life of the finned evaporator tubes. There was a 22% increase in the steam loading capacity of the HRSG.

5. Challenges faced

The major challenge faced with the new bellmouth and filters is the automatic puffing system to auto clean the filters became redundant. Even through the life of the new filters is long enough, they have to be replaced after they retire. The entire changing need the shutdown of the GT and the vacuum refrigeration cycle machine needs excess steam to run. There is also seen a coating of dust and oil accumulated in the chiller coil of the IAC filters and has to be cleaned to maintain the heat transfer. Running the chiller creates a little spike on the differential pressure of the filters and has to be monitored carefully with proper alarms. The filter also was additionally provided with blast windows which open automatically when the differential pressure reaches beyond a specific limit and was after the filters to make sure that the compressor never suffocates for inlet air. The HRSG also needed a complete changing of the tube structure in superheater II module. The control valves were tuned to feed additional water to compensate for the increase in evaporator tube bundle. Overall the entire project proved as a benefit to the efficiency and output and the limitations are feasible for the resultant output of the project.

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BIOGRAPHY

Utkarsh Adhyaru is a Mechanical Engineer and has been working in a captive power plant as a production and maintenance engineer. He has also been part of erection and commissioning of an 850 MW CPP and worked extensively in changing designs of boilers and burners. He also looks for the utilities of CPP and their process optimization.