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PROBLEM OF HIGH VIBRATIONS IN SERVO SYSTEM AT OVER LOADING CONDITION OF STEAM TURBINE

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ABSTRACT

Turbine governing is designed on the basis of power developed by the turbine on the same speed of shaft. When the power generation fluctuates from its designed value and governor not able to maintain its speed, plant operation becomes unreliable to the operators. So the maintenance of any governing system is a tedious job which requires very wise full decisions for its component overhauling and replacement, it requires vast practical knowledge and equipment behavior. After one start up of plant, taking a shutdown is very costly affair to afford huge cost of loss of primary content for a big process industry due to a very minute problem, is not easy to justify. After observing any problem in the system, it is not easy task to identify the correct cause of the problem because there are so many linking of the parameters to each other. Problem of hunting of control valves and hunting of speed is faced by many industries in the past years and now they have sufficient knowledge about it but high vibration in the servo motor and servo cylinders is new type for the plant people. Vibration level of 80-100 mm/s is a very high vibration and it is sufficient to break any joint of the lube/hydraulic oils and final cause of the turbine trip.

Keywords: HP& LP Valve- high pressure steam inlet valve & Low pressure steam inlet valve.

I. INTRODUCTION

For controlling the speed of the turbine governor is used and governor along with many systems is called governing system like servo motor, servo cylinder, spindle, amplifier etc and problem in any of these components become a cause of turbine stoppage.

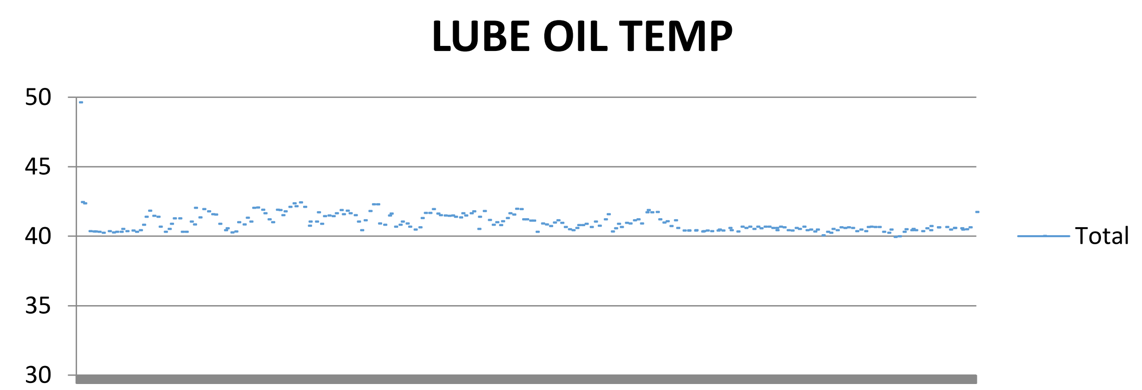
It is a pic of condensing come extraction type turbine, in which HP and LP valve is shown. Hp valve is for high pressure enters in the turbine and LP vale for the medium pressure entering in the second section of the turbine. Small up and down motion of the spindle is a normal phenomenon of the governing system but almost all turbines faced problem of high magnitude of up and down motion of spindle and causing speed fluctuation in the turbine. Certainly it is a known problem to the all plant operation people. It may be due to air locking in the governor or locking of rotation of the servomotor spindle (part no 17 of the servo motor) but very high vibration of servomotor and hydraulic is the new problem in the system which is generally faced after overloading of the turbine.

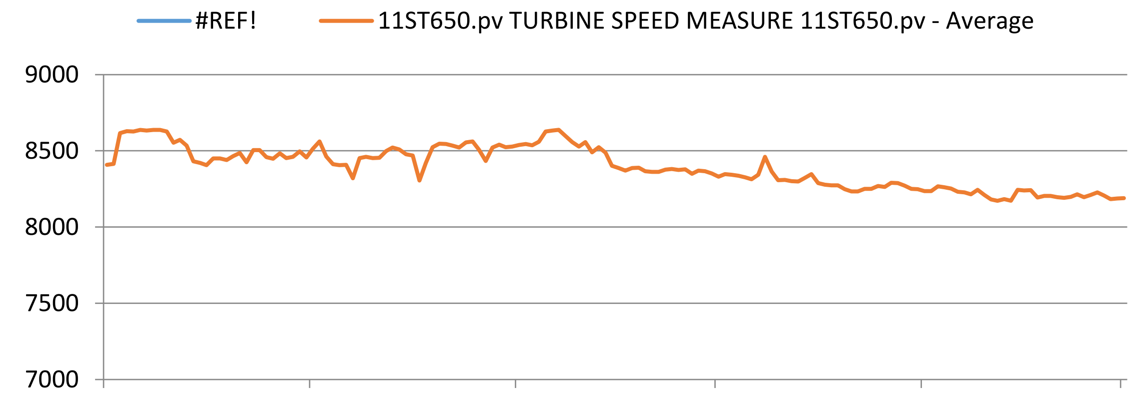
In this paper we will find problem cause and when such type of problem may come in picture. Vibration level (80-100mm/sec) of the servo motor and cylinder is so high that lines started cracking at the junction point of control oil, secondary oil lines and finally tripping of turbine. In this turbine LP servomotor was found in high vibration condition.

II. BACKGROUND OF THE EQUIPMENT

Turbine was installed in urea plant in 1994 for the compression of CO2 from 1kg to 160kg/cm2 and continuous running since inception without any problem. Since plant was designed initially for production of 2100 Ton per day production of urea and turbine was running trouble free but due to high demand of load plant load was increased continuously and after certain modifications plant is producing 3400Ton per day urea. So obviously turbine is highly overloaded.

Current running parameters-





High pressure steam flow (at 105kg/cm2)-73T/H
Condensing load-15.5T/H by measuring flow at the condensate outlet
Extraction steam flow=73-15.5=57.5T/H (back calculation done due to faulty flow transmitter of extraction steam)

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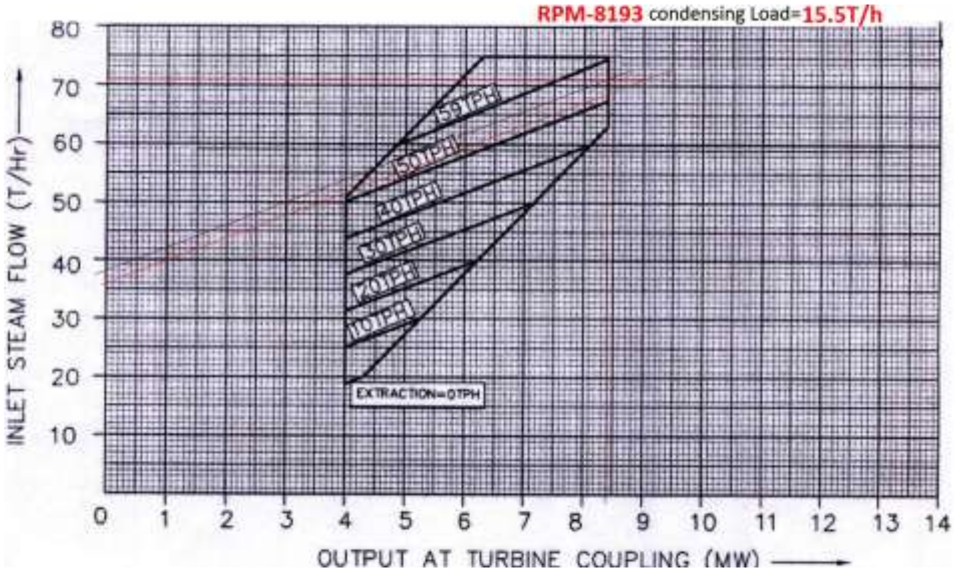
Design parameters-

TURBINE DESIGN DATA

TYPE: EHNK 32/29-3 — SHT. 01 OF 01 SHTS.

		UNITS
OUTPUT		
MAXIMUM OUTPUT	8400	KW
DESIGN RATING (ECONOMICAL RATING)	8400	KW
SPEED		
SPEED TURBINE (5)	8029	rpm
SPEED REDUCTION GEAR OUTPUT SHAFT	-	rpm
SPEED RANGE	7290~8505	rpm
PRESSURE		
SPECIFIED INITIAL STEAM PRESSURE (5)	104.0	Kg/cm2(Abs)
PERMISSIBLE DEVIATION WITHOUT LIMITATION (1)	110.9	Kg/cm2(Abs)
PERMISSIBLE DEVIATION (2)	115.9	Kg/cm2(Abs)
PERMISSIBLE DEVIATION INSTANTANEOUSLY FOR A TOTAL DURATION OF 12 HOURS PER ANNUM (2)	138.0	Kg/cm2(Abs)
PRESSURES INSIDE THE TURBINE		
THE PRESSURE IN THE H.P. WHEEL CHAMBER OF THE TURBINE MUST NOT EXCEED THE FOLLOWING VALUE AT MAXIMUM LOAD AND WITH SALT DEPOSITS ON THE BLADING :	77.0	Kg/cm2(Abs)
THE PRESSURE IN THE I.P. WHEEL CHAMBER OF THE TURBINE MUST NOT EXCEED THR FOLLOWING VALUE :	-	
EXTRACTION PRESSURE (5)	26.5	Kg/cm2(Abs)
THE EXTRACTION PRESSURE MAY RISE TO	27.5	Kg/cm2(Abs)
AND DROP TO	25.5	Kg/cm2(Abs)
PRESSURE AT EXHAUST FLANGE (5)	0.12	Kg/cm2(Abs)
TEMPERATURES		
SPECIFIED INITIAL STEAM TEMPERATURE (5)	504.0	°C
PERMISSIBLE DEVIATION WITHOUT LIMITATION (1)	510.0	°C
PERMISSIBLE DEVIATION FOR LONGER PERIODS (2)	518.0	°C
PERMISSIBLE DEVIATION FOR 400 HOURS PER ANNUM (2)	524.0	°C
PERMISSIBLE DEVIATION FOR 80 HOURS PER ANNUM (2)	538.0	°C
1. ANNUAL MEAN VALUE. 2. THE ANNUAL MEAN VALUE (1) MUST NOT EXCEEDED. 3. THE TEMPERATURES AND PRESSURES WHICH CAN BE MAINTAINED FOR LIMITED PERIODS ONLY ARE PERMISSIBLE FOR UNFORESEEN VARIATIONS IN THE OPERATING CONDITIONS. IT BEING UNDERSTOOD THAT OPERATION AT THESE VALUES WILL BE WITHIN NARROW LIMITS, IN PARTICULAR WHEN MAXIMUM TEMPERATURE AND MAXIMUM PRESSURE OCCUR AT THE SAME TIME. 4. THE FIGURES INDICATED ARE LIMIT VALUES. IT IS ADVISABLE TO USE LONGER PERIODS TO KEEP DOWN THE STESSES IN THE MATERIAL. 5. DESIGN		
COOLING AIR TEMPERATURE (CONDENSER)	35.0	°C

STEAM & GAS TURBINES AND COMPRESSORS — DEPT. T&C Engg. — DEPT. CODE 415



III. ANALYSIS OF THE PROBLEM

From above Specification sheet we find that turbine is design for maximum output 8.4MW and speed range 7290 to8505. If we go through running condition of the turbine we find that turbine is running at its maximum load (called overloading).

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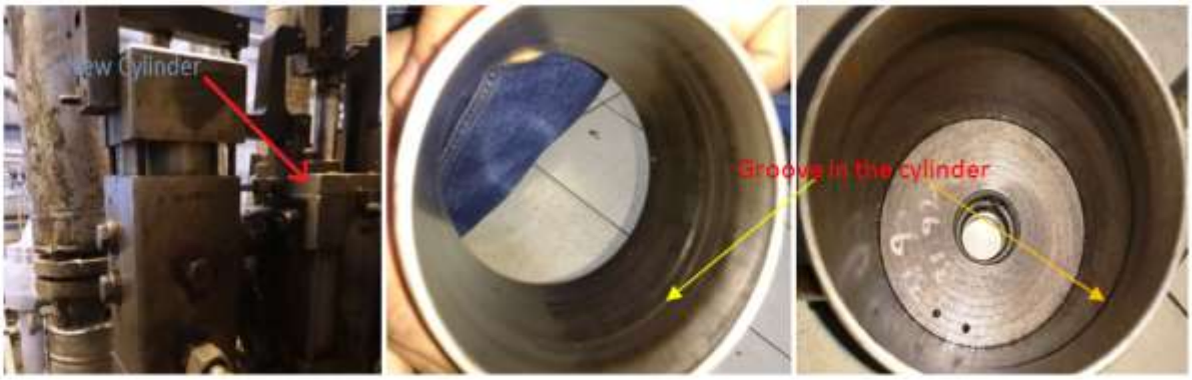
From above curve (supplied by manufacturer), turbine is generating power approx 8.3Mw which is its maximum output hence turbine is running at its overloading condition (there should be always a 5-10% margin from its max. output).

Maximum RPM is 8505 (over speed trip value) and turbine always should below 10% of max but in this case it is 0.03%. So it is clear that turbine is running at overloading condition which is not recommended by any manufacturer.

In this case, governor was overhauled and calibrated and fitted 4 months before this problem appearing so there was no chance to have a doubt in governor. In running turbine there are very few options which may be tried to get the cause. So finding the cause of fluctuation in the LP extraction valve opening we changed the extraction steam flow by giving command to Cramer device but no change in the fluctuation occurred. There are two extraction steam valve so we reduce the opening of valve one by one up to its minimum and increase the extraction from second valve. By doing this it is found that problem is resolved by reducing the opening of left hand side valve. This was the method by which we identify the problem in left side of servo system. So when we got the opportunity, open and dismantle the servo motor and cylinder, find that deep groove inside the cylinder (deep scoring), so replaced the cylinder. We can see it more clearly in the pictures. After dismantling of the hydraulic cylinder it is seen that the problem was due to wear of the cylinder and hydraulic oil by passing through piston oil seals. Cylinder pics are –



At bottom of the cylinder hydraulic pressure is around 8kg/cm2 and after crossing piston is pressure is 2kg/cm2 (line back pressure) connected to drain.
P1-P2=6kg/cm2

By energy balancing
$(P1-P2)/\rho=(V1^2-V2^2)/2/g$ where ρ is lube oil density
ρ for the Oil will change as per below equation
$\rho 1 = m / (V0 (1 - \beta (t1 - t0))$ Volumetric Temperature Coefficients – β

Since Lube oil is a petroleum product

β for Petroleum products	0.0010°C

Density of T-46 grade oil-861kg/m3 at 40°C
Temperature inside the cylinder is approx 100°C
$\rho 1 = m / (V0 (1 - \beta (t1 - t0))$ *=861(1+.001(100-40))=344kg/m3*
Pressure inside the cylinder=8kg/cm2=80000kg/m2

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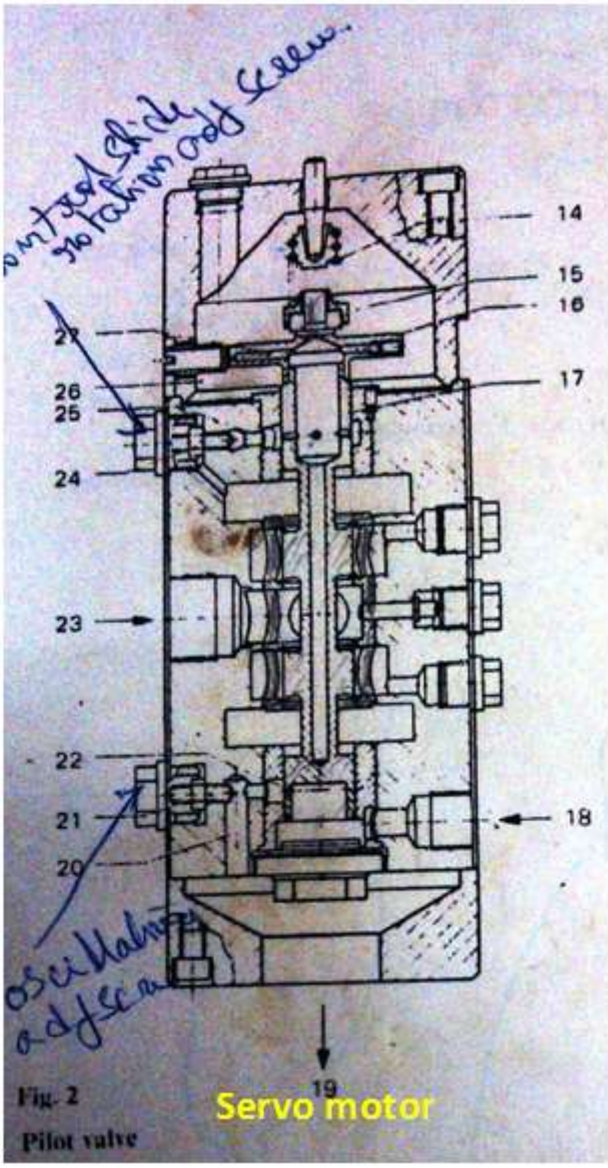

Fig. 2
Pilot valve

Assuming the initial velocity zero as the fluid which will come in contact with the piston will be stagnant so
V1=0 m/s & P2=2kg/cm2 (connected to the drain but considering pipe resistance)
$V2^2=2g*(P1-P2)/\rho$
$=2*9.8*(8-2)*10000/344$
V2 =58.46m/s this the velocity inside the cylinder if two phase flow is not taking place

Normal velocity inside any cylinder (considering cylinder as a pipe) is not recommended more than 3m/s and above 3m/s turbulence is started and it is a tremendous large value for creating the turbulence, magnitude of turbulence will also be very high and due to which complete servo system was vibrating.
So this type of vibrations occurs when the fluid passes the piston seals from one side to the other (high pressure to drain side).

Why & when this problem may occur-

1. After dismantling of the cylinder it is seen that cylinder is scored from one side (180degree) only and other side there is no groove.
2. Groove is at the top of cylinder.

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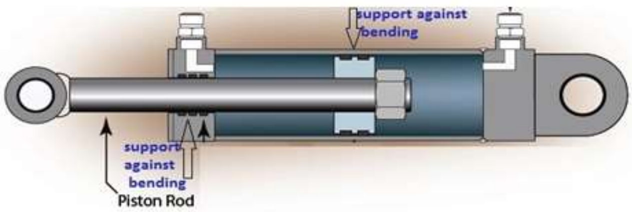
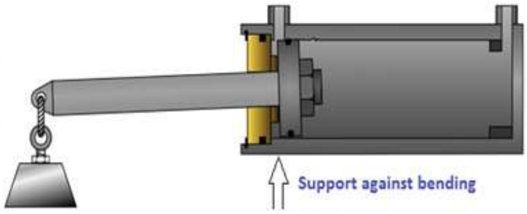
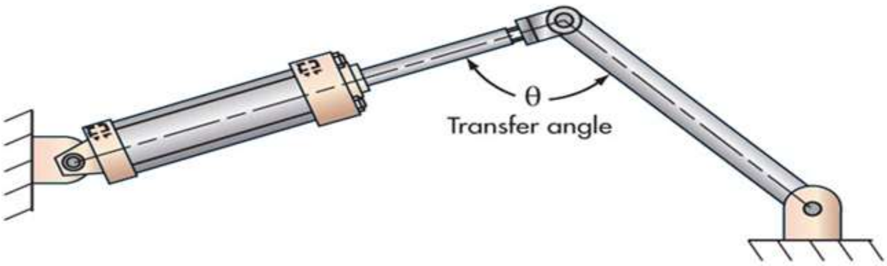
On turbine overloading condition valve will be in full open condition and also above both observation states same. In top position of piston load cannot be taken care by piston due to improper supporting (supporting become from two different positions (one at middle and at cap) to one position (only near cap)) and increasing bending movement of piston rod.







Cylinder is hinged at one position so it is free to move in its pin but due to high axial force of spring on pin and wear of pin, it will always creates some frictional resistance and restrict its rotation which is sufficient to create such type groove after running of a certain duration.

IV. CONCLUSION

In overloading condition of the turbine all actuators operate in its maximum opening condition and forms groove which become a cause of the bypassing of the piston fluid flow from high pressure zone to low pressure zone which creates turbulence inside the cylinder, this turbulence creates vibrations in the cylinder and cause the final breakdown of the turbine. When fluid bypasses the piston so governor takes action and supplies some more fluid to the cylinder to maintain its position and it continues which also contribute in high vibration in the system and also fluctuate speed up to some extent.

If possible not to overload the turbine than it is best; otherwise always check pin and piston condition after running of 4-5 years of operation and as per observation parts should be replaced as a best practice

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