A VIBRATION ANALYSIS OF A TURBO-GENERATOR DEFECT

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Methods of vibration measurement and analysis for machinery health were described in a recent article[1] in the *Journal*. The majority of surveys undertaken by the Fleet Vibration Analysis Unit are of a routine nature and therefore pre-planned. There are however occasions when, on request, a member of the team is despatched to a ship in order to assist in the identification of a particular machine defect. The following is an account of the follow-up to just such a request.

The machine in question was a turbo-alternator in a LEANDER Class frigate, which at the time was working up at Portland.

Before joining the ship the results of any previous surveys of that particular machine are studied, along with the relevant machinery information docket, which tabulates any predictable frequencies associated with the machine, e.g. running speeds, gear meshing frequencies, etc.

On arrival on board, discussions with ship's staff are of course very important, and details of just how and when the problem arose can be helpful. In this particular case the vibrations generated were most noticeably felt through the watchkeepers' feet on surrounding structures and deckplates. Ship's staff had 'felt' around the generator and eventually decided that the most severe vibration was at the generator bearing, and they therefore suspected that this was the source of the problem. In the event this prognosis proved to be incorrect, which demonstrates the difficulty of pin-pointing any defect due to vibration without the use of an instrument capable of discriminating between various frequencies. Vibration can often be transmitted through a machine or supporting structure so that the most severe vibration may be experienced at a position far removed from the source of the defect.

The survey was conducted using IRD 820 and 880 vibration analysers, with a model 970 accelerometer transducer. Readings were taken at the positions indicated on the outline machine sketch in FIG. 1.

On commencing the survey it soon became apparent that the predominant frequencies being generated throughout the machine were due to an increase of some 50% in the amplitudes recorded at a previous survey at turbine fundamental frequency and the 'appearance' of vibration at approximately half turbine speed. These amplitudes were present throughout the machine but only those recorded on the high speed line itself (A,B,C, and D) are reproduced in FIG. 2.

The half speed frequency was the more significant. Whereas a level of vibration will always be detected at the rotational speed of the machine, amplitudes at half speed would not usually be expected and these were thought to be created by oil whirl. Oil whirl is known only to occur in machines equipped with pressure lubricated sleeve bearings and operating at relatively high speeds. It occurs at between 3% and 8% below half speed of the machine.

The mechanism of oil whirl is best described as follows. Under normal operating conditions the journal of a rotating machine will ride up the side of the bearing, drawing the pressurized lubricating oil into a wedge to

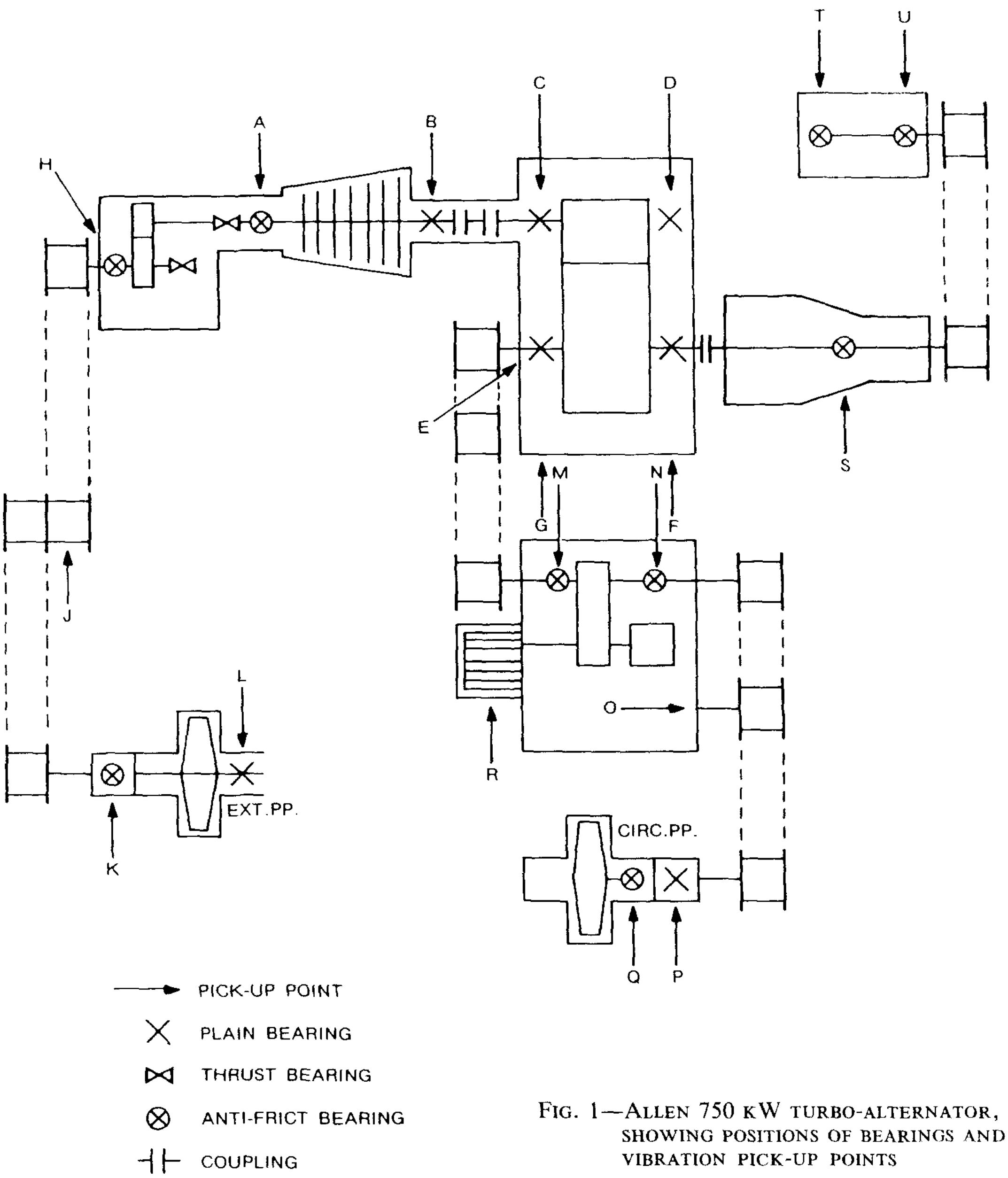

FIG. 1—ALLEN 750 kW TURBO-ALTERNATOR, SHOWING POSITIONS OF BEARINGS AND VIBRATION PICK-UP POINTS

produce a load-carrying film. Molecules of oil adjacent to the journal will tend to stick to it and therefore rotate at machine r.p.m.; the oil molecules in contact with the stationary bearing adhere to it and do not rotate. As a result the oil between the journal and bearing is in shear and tends to rotate at a speed which is the average of the shaft and bearing r.p.m. This is half of turbine speed, less 3%–8% due to fiction losses. It should be noted that pressure-lubricated bearings will always contain oil rotating at half machine speed, but under normal circumstances the resulting shear forces are very small and would not be expected to be significant. Should the internal clearances within the bearing become excessive, however, the forces due to the rotating oil film can predominate and in these circumstances the journal can be caused to whirl around the internal surfaces of the bearing. This is oil whirl.

AMPLITUDE AND FREQUENCY DUE TO TURBINE FUNDAMENTAL

AMPLITUDE AND FREQUENCY DUE TO OIL WHIRL

PICK-UP POSITION		FILTER OUT		AMPLITUDE AND FREQUENCY DUE TO OIL WHIRL		AMPLITUDE AND FREQUENCY DUE TO TURBINE FUNDAMENTAL	
		S/Eg	mm/s	mm/s	cpm	mm/s	cpm
A	H		8	3.5	5.6k	4.5	12k
	V		3.5	1		1.8	
	Ax		2.5	1.2		0.8	
B	H		3.5	3		1.2	
	V		2.2				
	Ax		2.1				
C	H		3.5	1.5			
	V		3	1.8		0.5	
	Ax		5	3.6		1.2	
D	H		3	1.2			
	V		1.9				
	Ax		3.5	3.2		0.8	

FIG. 2—VIBRATION LEVELS RECORDED BEFORE RENEWAL OF TURBINE BEARINGS
rotational speed of turbine: 12 000 r.p.m.

AMPLITUDE AND FREQUENCY DUE TO TURBINE FUNDAMENTAL RETURNED TO NORMAL LEVELS

COMPLETE DISAPPEARANCE OF AMPLITUDES DUE TO OIL WHIRL

PICK-UP POSITION		FILTER OUT		COMPLETE DISAPPEARANCE OF AMPLITUDES DUE TO OIL WHIRL		AMPLITUDE AND FREQUENCY DUE TO TURBINE FUNDAMENTAL RETURNED TO NORMAL LEVELS	
		S/Eg	mm/s	mm/s	cpm	mm/s	cpm
A	H		2.6	NIL	5.6k	1.3	12k
	V		3.2			1	
	Ax		2.2				
B	H		2.2				
	V		1.5				
	Ax		3.4			0.4	
C	H		3.5				
	V		3.5				
	Ax		4.5			1	
D	H		3.8				
	V		4.4				
	Ax		2.2				

FIG. 3—VIBRATION LEVELS RECORDED AFTER RENEWAL OF TURBINE BEARINGS
rotational speed of turbine: 12 000 r.p.m.

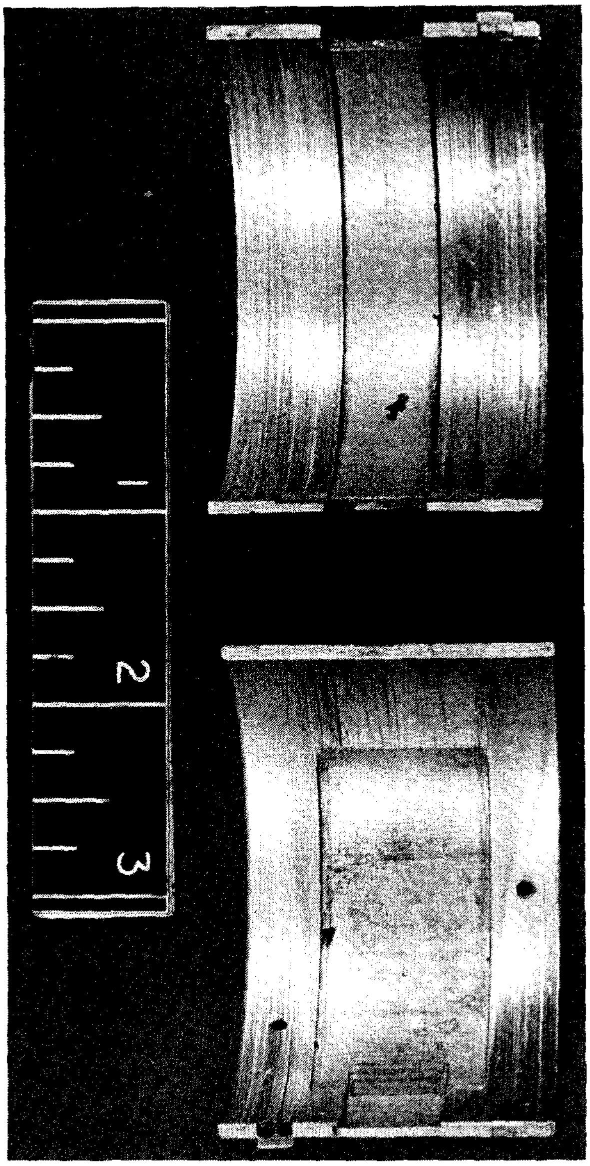

FIG. 4—WORN TURBINE FREE END BEARING (POINT A) FROM ALLEN 750 kW TURBO-ALTERNATOR

Having thus established a possible cause of the defect as being worn turbine bearings, it was worthwhile, before completing the survey, to attempt to prove that the frequency in question was due to oil whirl. Noting that the oil pressure wedge is pushing the shaft around, if the oil condition itself could be altered (i.e. pressure or temperature), there would be a resulting effect on the rate at which the journal is moved within the bearing. As the oil pressure on a running turbo-alternator cannot be altered it was decided to alter the oil temperature. This was done and the oil temperature from the cooler was reduced from 120°F to 95°F, thus increasing the oil viscosity and therefore increasing its push effect. As the oil temperature was reduced the frequency of vibration caused by the oil whirl increased and the opposite effect occurred when the oil temperature was returned to normal. This experiment proved that the oil condition had a direct effect on the source of the predominant vibration which had first attracted the attention of ship's staff.

The ship was recommended to inspect the turbine bearings at the first opportunity. This was subsequently carried out and the bearings were found to be excessively worn (FIG. 4). Poker gauge readings show that the turbine shaft was lifted 0.009 inches at the free end and 0.0065 inches at the driven end once new bearings were fitted. The bearing removed from the driven end of the turbine was mislaid so FIG. 4 shows only the one from the free end. Ship's staff decided to replace the pinion bearings at positions C and D as well as the turbine bearings.

A post-repair VA survey was carried out after the repair (FIG. 3). All vibration amplitudes due to oil whirl had disappeared and vibration occurring at the turbine fundamental frequency had returned to normal.

References

1. Thorne, R. W.: Good vibrations—bad vibrations; *Journal of Naval Engineering*, vol. 29, no. 2, June 1985, pp. 147-153.