# Soundcard Based Dynamic Balancing Machine for Model Gas Turbine

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## ABSTRACT

*Balancing of the rotor is one of the most important crucial key elements of the gas turbine construction. In this article the author will present a new method for balancing model gas turbine. The main aim was to implement a sophisticated balancing machine for homebuilders without large investment, complicated electronics and mechanics. Every balancing machine consists of two parts: mechanics and electronics and/or software. The design concept of the mechanical part is to provide easy balancing of complete turbine. This in situ, soft suspension, balancing system (balancing in own support) helps to correct the unbalance problems coming from bearings and shaft misalignments (angular and parallel). The other advantage of this system is implementing the balancing instrumentation virtually without electronics. Only an electronic switch provides the selection between left and right signal of the sensors. All measurement functions are implemented by software with digital signal processing, average workshop personal computer and low cost soundcard. This project has not completely finished yet, but it can probably help for other gas turbine enthusiasts all over the world.*

***Keywords**: dynamic balancing, soft suspension, in situ balancing, gas turbine, jet engine, soundcard, RC, digital signal processing, dynamic unbalance, static unbalance, couple unbalance*

## INTRODUCTION

Balancing of the rotor is one of the most important key elements of the gas turbine construction. If the rotor unbalance grade is G0.4 (ISO 1940/1 the strictest grade for gyroscopes) and the rotor weight is 0.5kg, it can causes 0.017gmm unbalance (the grade relative to the weight of rotor and to the revolution speed).

$$U[gmm] = 9549 \cdot G \cdot M[kg] / n$$

**Equation 1 Maximum allowable unbalance calculation from grade**

It means that there is a 0.017g weight at 1mm radius (but its RPM out of the maximum service speed range of the standard). This unbalance makes the following force:

$$F = mr(2\pi n / 60)^2$$

**Equation 2 Unbalance caused force**

The value of the force generated by eccentricity in the example it is 2.5N at 117000RPM. This unbalance should be split into two planes according to the mass distribution of the shaft. This relatively high force effects on the bearings and the bearing reacts to the main shaft. It can cause vibration, shorter lifetime (fatigue), deformation, power degradation, fraction and can be dangerous for life, despite of the fact that this is the lowest grade from the standard. **There is no exception the precise balancing has to be done for every gas turbine.**

In the case of rotating shaft the unbalance causes periodical forces to the suspension and the periodicity corresponds to the rotational speed or with other words it is synchronous with rotational speed (first order). In order to balance the rotor the vibration of shaft revolution frequency should be selectable in the balancing instrument. This reduces the disturbance caused by the noise, harmonics, bearings and blade frequencies etc.

The unbalance is radial in their line of action and it is a vector quantity. It has both size and direction. The direction can be characterised by the phase between the unbalance vector (from the centre of the shaft) and a vector to the reference point at the shaft (from the centre of the shaft).

The general dynamic unbalance consists of the static (single plane unbalance) and couple unbalance. The former is when the mass centre line is not the same but parallel with the rotational axis. Only this kind of balance exists in disk shape structures. It can be eliminated by only one compensating weight.

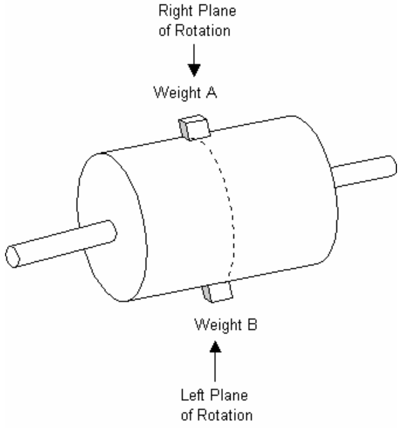


**Figure 1 Static unbalance**

The later is when a pair of weight is at the two ends of the shaft but opposite side of one to other (180°). The rotor is in static balance, but the centrifugal forces will produce a moment about the centre of mass when the rotor turns. In the case when only couple unbalance exist the mass centre line cross the shaft axes at the mass centre point.

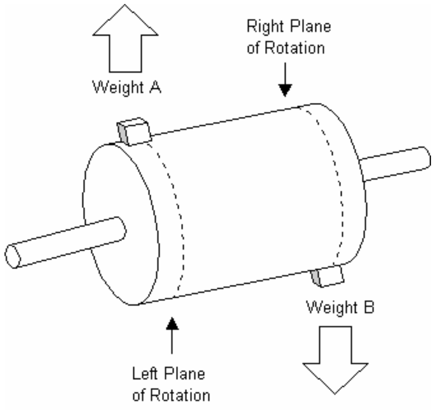


**Figure 2 Couple unbalance**[1]

[1] http://www.balancemaster.com/Balancing_explained.htm

The couple unbalance can be compensated by two weights, which were put to counteract the couple unbalance at two planes.
The ideal balancing task is to reduce the inhomogeneous mass distribution caused forces by putting or removing weights along the shaft, but the most cases are enough to use two or more correction planes. In the balancing system the vibration produced by unbalance are measured at two planes and after the compensation weight and place are calculated to the correction plane from the measurement. Unfortunately the unbalance at the one side can cause shaft movement around the mass centre point of the rotor and it makes movement at the other side. These phenomena called cross effect, because the unbalance at one side can not be treated independently from the other side.
In this article simple software based balancing system for model gas turbine including mechanics will be described. The mechanics is as so important part as the electronics and software. The most sophisticated signal processing method without reliable and precise mechanics can not achieve proper result. Therefore the first part of the article will be about the mechanics and the design rules for the sensors and the second part will be about the PC software, its advantages and further possible enhancements.

## LITTLE HISTORY

**János Horváth** initiated and supports this project. János has a very good machinery factory and he is a "ancient" modeller since 1969. He started his hobby with helicopters and he has seen jet engine for 3-4 years, and at the first sight he has fallen in love with it. He has built by his own a KJ-66, and put into a small ALBATROS. Currently, he is constructing a GRIPEN 1:5.5, and he wanted to build a 120-160N trust jet engine for it. We started to work together to tune and enhance his Phoenix Mk-4 and the FADEC. The first, should be solved thing, was the balancing of the turbine (In my viewpoint, it is very important basic element of GT construction). I built the electronics for the balancing machine, but we were not satisfied with the result. It has ambiguous readings (power LED strobe) at the low level of unbalance. (In my opinion the original mechanics works near the resonance frequency; it is nor a hard and neither soft suspensions machine). I decided we should improve the capabilities of the balancer. Currently we have an own in situ (complete turbine), DSP based balancing machine, but we have to do lot of things to enhance the machine and to try them.
János and his friends organise the World Masters in 2005. We believe in that we can learn from our friend all over the world including members of GTBA. Maybe World Master will increase the popularity of the GT technology in Hungary. Nowadays, only few people build GT propelled aeroplane in Hungary and I think that János is a leader in building own GT engine.

## THE SENSORS

Two kinds of sensor are used in the balancing machine what convert the mechanical movement and the position information to electrical signal. The first is the vibration sensor and the second is the position reference sensor.

### *Vibration sensors*

Sony® 80mm in diameter loudspeakers with 50mm magnet in diameter was selected for the vibration sensors. The sensor rod should be glued with epoxy resin to the membrane of speaker. It has more advantages than disadvantages for this application.

Advantages of using loudspeakers as vibration sensor in balancing machine:

- Low impedance, especially at low frequency range (where they are used in balancing machine).
- Low sensitivity for the hum (50 and 60Hz), it is consequence of low impedance.
- Velocity sensitive sensor (U ~ v).
- Low high frequency sensitivity vs. accelerometer. It comes from acceleration is the differential of velocity. This means that it suppress the high frequencies noise.

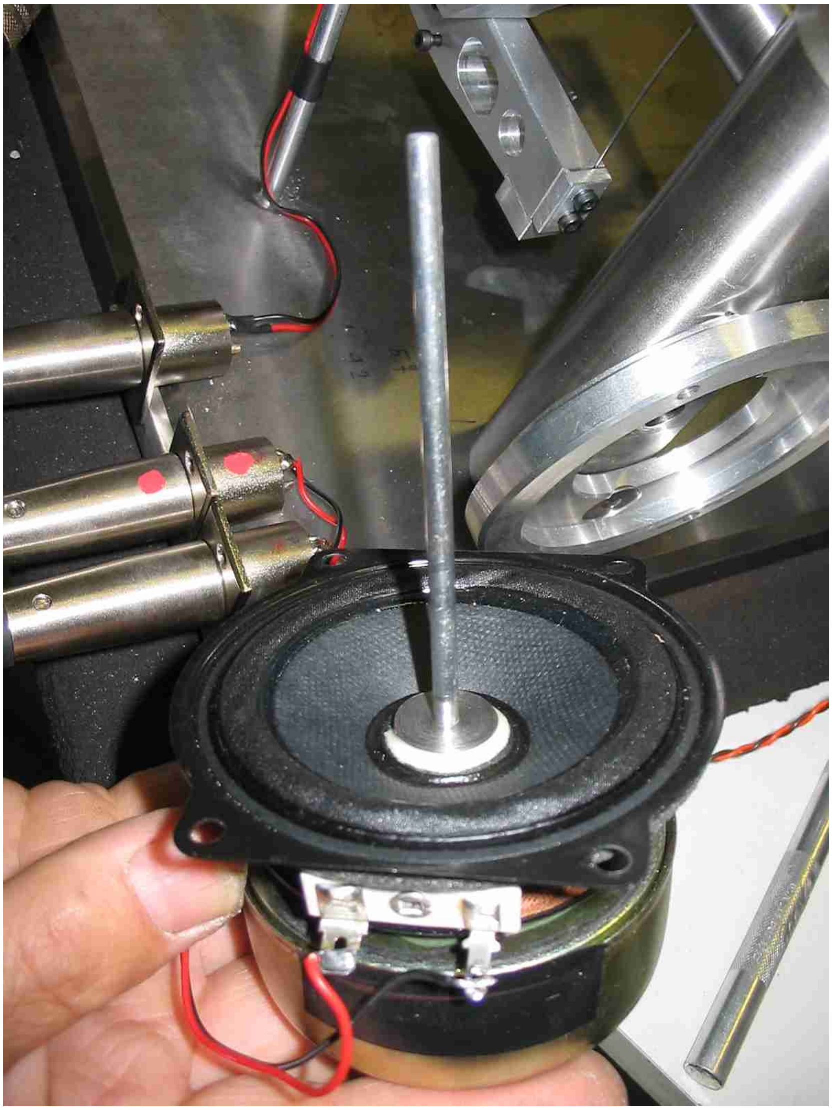

**Figure 3 Using loudspeaker as a vibration velocity sensor**

- It is relatively soft suspension; complete mechanics with turbine has low resonance frequency. The balancer is used above its resonance frequency.
- It does not require signal conditioning and power supply; it has floating output (independent from earth to avoid earth loops).

- Low cost, easily replaceable.
- Not fragile.
- It has not high temperature dependency (Piezo has).
- Free angular movement, about ±20°.
- Off-the-shelf available.

Disadvantages of loudspeakers:

- Low output voltage vs. piezo sensors.
- High sound (large surface, microphone) and ambient or base vibration sensitivity.
- Membrane is very volatile, and the rubber ring is an ageing component.
- Over the free angular range it can squeeze and cause phase problems if it is small degree, or amplitude and phase problems if it is large.

**Figure 4 Fixing of the speakers**

### *Position reference sensor*

A coil of disassembled relay (12V) was chosen for position reference sensor.
Advantages of using inductive revolution sensor for balancing machine:

- Relatively low impedance, especially at low frequency range (where it is used in balancing machine)
- It does not require signal conditioning and power supply; it has floating output (independent from earth to avoid earth loops).
- It has pure sine-wave signal output.
- It provides contactless measurement.
- It uses the built-in magnet ring of the rotor.
- It does not have hysteresis.
- It is not disturbed by ambient light.
- It is very simple and cheap.
- Changing the distance between the magnet and the sensor can control the signal level.

Disadvantages of using inductive revolution sensor:

- The exact phase of the signal cannot be known from the magnet position, it should be determined indirectly.
- It is sensitive for external magnetic field including mains field (hum, high number of turns).

A non-magnetic material holder should mount this sensor. Please, do not ground any wire at the sensor side to avoid earth loop, but all metal parts of the balancing machine must wired to protective earth of the mains (ESD and life protection).

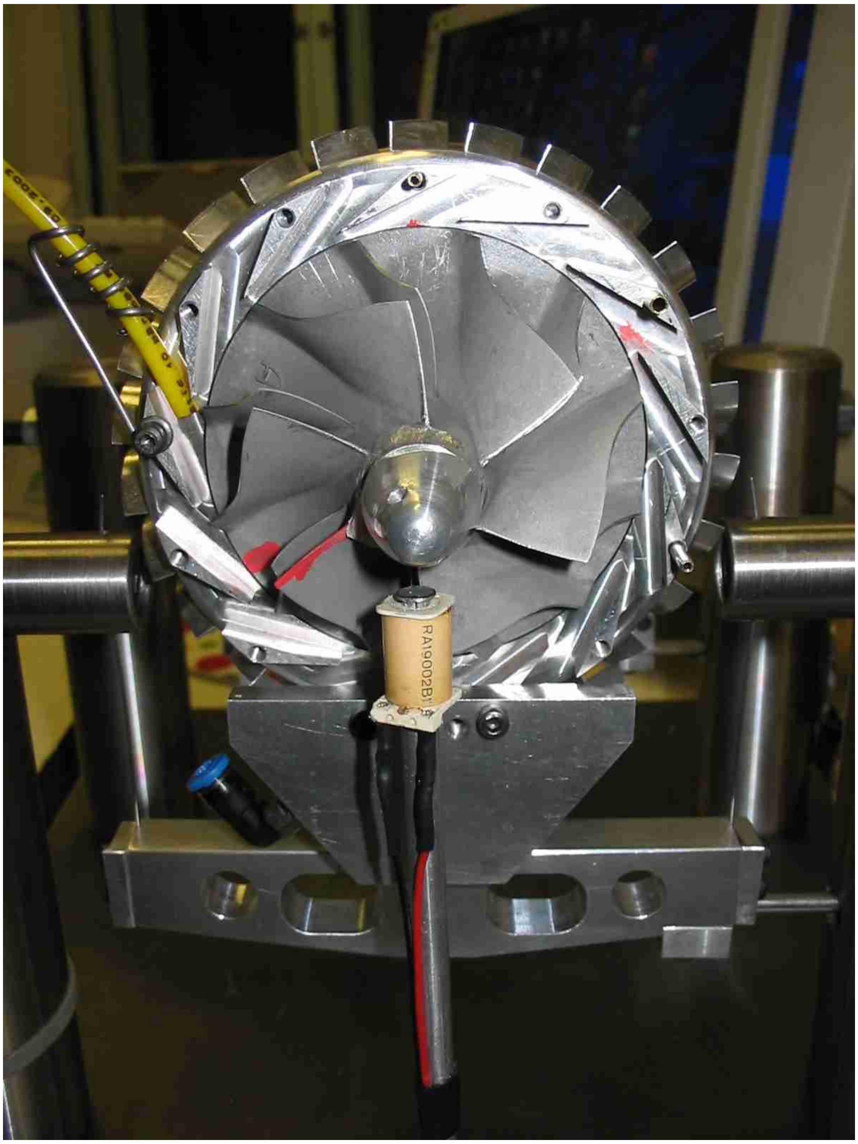


**Figure 5 Inductive revolution sensor**

## THE BALANCING MECHANICS

The following design considerations for the balancing mechanics should be taken into account:

- It should have low resonance frequency to minimise the phase error at the measurement revolution (the measurement revolution is above the self-frequency of mass and spring system consists of the sensor and turbine).
- Soft suspension construction to provide the low resonance frequency.

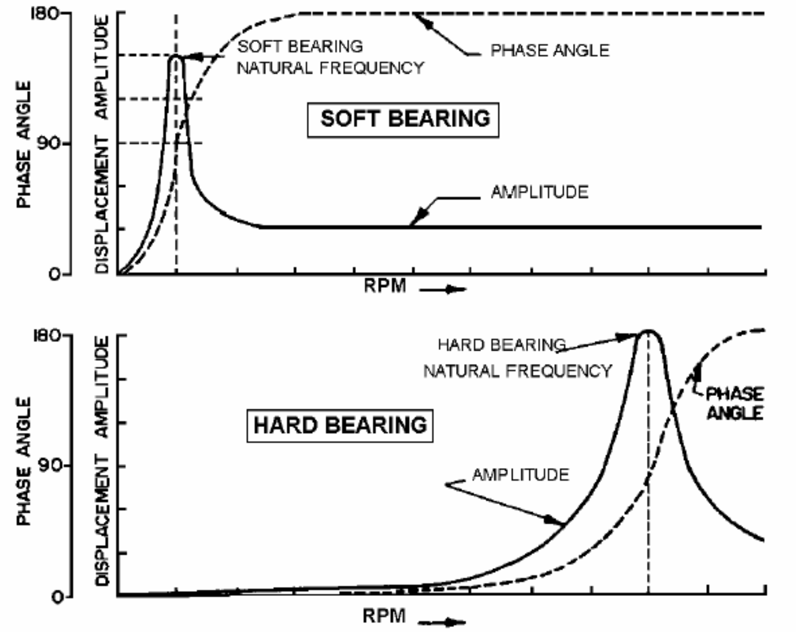


**Figure 6 Soft suspension vs. hard suspension[2]**

- Hold the whole turbine with solid fixing without flexible and loose connection to the sensor system. (It can cause phase and amplitude uncertainty).
- It has to allow the horizontal turn of the whole support around its mass centre point.
- It has to allow the movement of whole support parallel with sensors.
- It has to allow the length change, what is consequence of traverse movement.
- The spring force against movement should be same for parallel and opposite directions.
- It should keep the sensor system stable; it has to have large mass base to reduce the external vibration sensitivity.
- It has a fixing point for compressed air tube to the compressor wheel.
- Hold the vibration and rotation sensors.
- It provides the lubrication of the bearings.
- It should be mechanically isolated from the bench to avoid the disturbance of the conducted vibrations (a foam sheet and the mass of the mechanics form a mechanical low pass filter and it de-couples the vibration of the bench)

[2] 14. RYAN, Vic, Complete Site Maintenance (CSM) The Way Forward, Electrical Repairers' Convention - '96, 13th - 15th September, 1996 Mount Amanzi Lodge, Hartebeespoort Dam

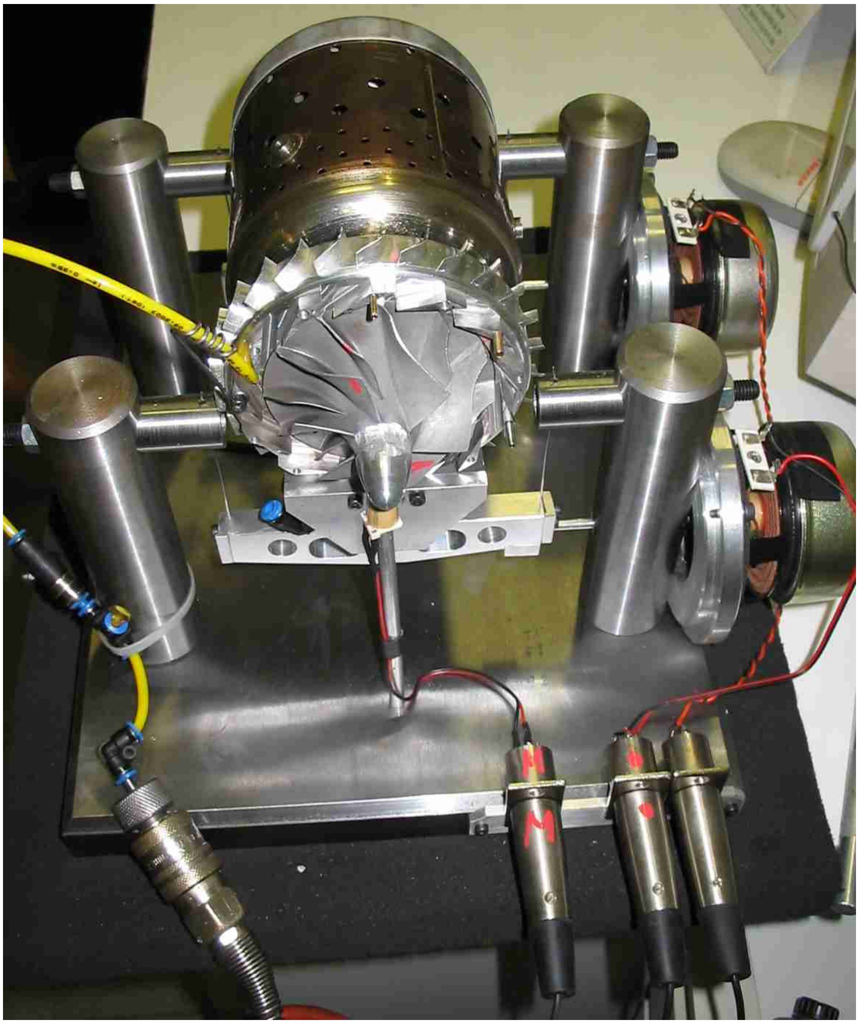

**Figure 7 Complete balancing machine with Phoenix Mk-4**

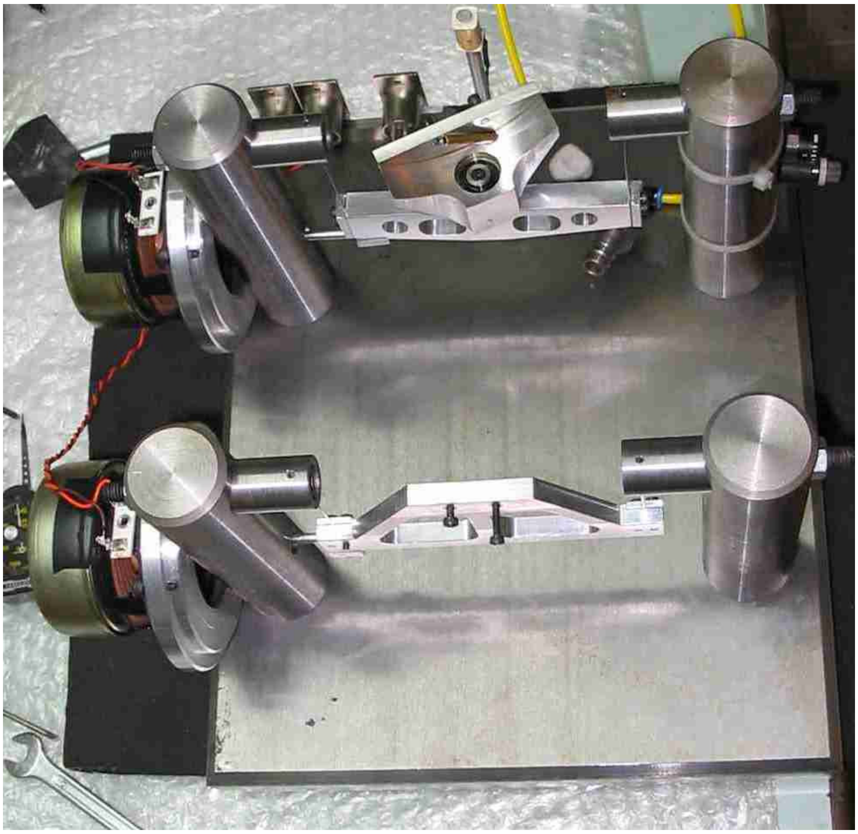

**Figure 8 Top view of the complete balancing machine without turbine**

**Figure 9 Rear view of balancing machine with turbine**

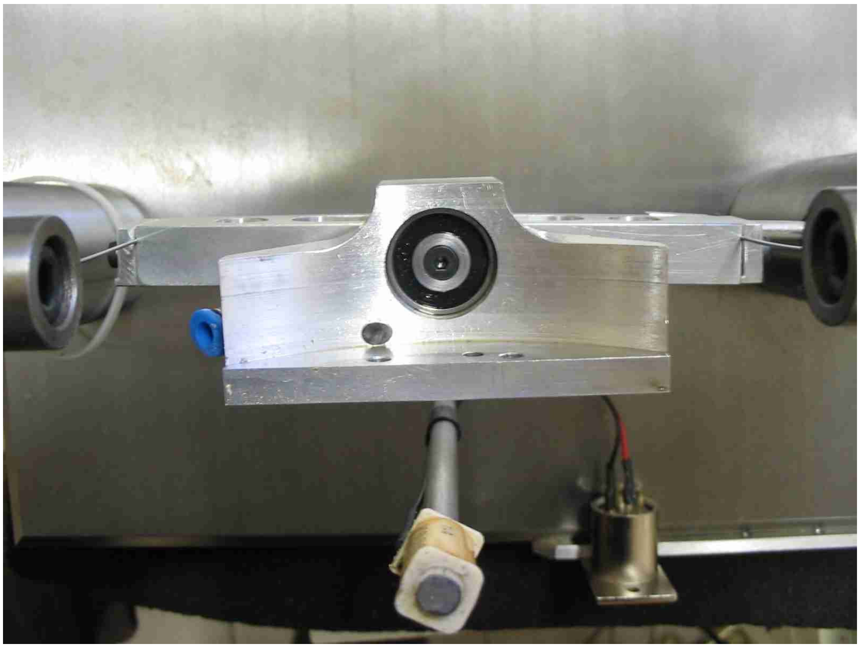

**Figure 10 Traverse bearings**

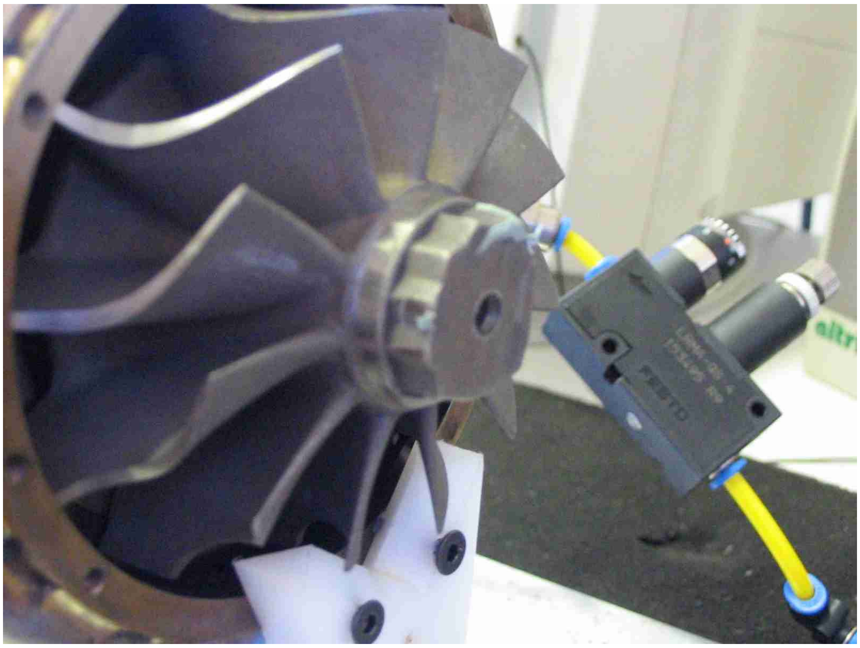

**Figure 11 Compressed air pressure regulator**

The pictures about the balancer tell everything, how the mechanics was implemented. The whole turbine hangs on 0.8mm in diameter 50mm long steel wires and these allow the free movement of the all support. The base-plate should be a very heavy sheet and the all system mounted to it. The rod of the speakers is directly screwed with a plate to the support in order to transfer the vibration to the sensors. The wheel was driven at the compressor side by controlled pressure compressed air. The traverse bearings allow the cross direction movement, but it probably can be omitted, because the wire springs allow this kind of movement (It should be try!). The reference position sensor is built to the base plate with an aluminium rod and the posts keep the speakers. The turbine is screwed to the swing with hex nut bolts, but it not a universal mounting method, it is currently good only for Phoenix Mk-4.

## THE ELECTRONICS

The one of the advantage of described method is the simple electronics, what is not necessary to build, but it makes the balancing task simpler and quicker. The electronics has only one duty to switch between left and right side signal of the vibration sensors. It connected to a free parallel port of the PC and contains only a high quality shielded reed relay with built in protection diode. It fits a simple sub-D connector house and the reed relay is wired to a 25 pole male connector. The schematic is under the text.

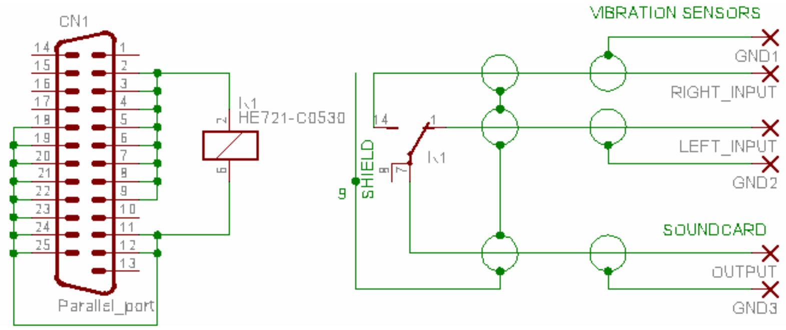


**Figure 12 Vibration sensor selector switch schematic**

The audio wiring has to be done by shielded cable. At the soundcard end a 3,5mm in diameter jack connector and the sensors end RCA connectors are recommended. Use insulated socket at the sensor side and all ground should be isolated from the metal parts and the PC earth to avoid ground loops. The soundcard left channel input (white) should be directly connected to position reference sensor and the switch output should be connected to soundcard right input (red). The used parallel port can be set in the program (LPT1:, LPT2:).
If you omit this electronics you have to change between the sides by the change of the connectors.

## THE BALANCING SOFTWARE

Nowadays, average home PC's have internal soundcard and their computational power allows to use them even in real time signal processing applications. Using soundcard as analogue-digital converter very high performance test instruments can be implemented without additional electronics. CraetiveLabs® Audigy® and Audigy2® or similar recommended, because they have good quality, high signal-noise ration and low distortion, but any other type of soundcards is carefully usable (the minimal sensible unbalance may be bigger, or the result can be worse).

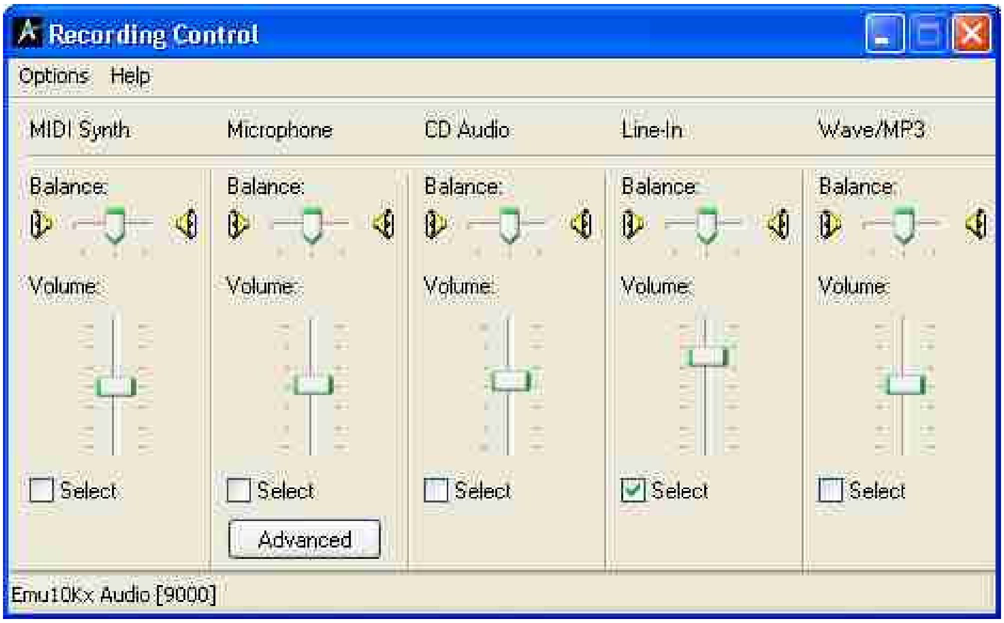


**Figure 13 Soundcard mixer settings**

The line input of the soundcard has to be selected, and the mixer should be set around ¾ of the maximum position. This is the optimum point, where the gain is relatively high, the distortion and the noise is relatively low, but it depends on the card and it requires experimentation. All measurement functions are implemented by software and displayed on the screen.

**Figure 14 CreativeLabs® Audigy® connectors**

Recommended minimum system requirements:

- Processor: Celeron 850MHz
- RAM: 64Mbyte
- Video: SVGA 1024x768
- Ports: 1 parallel
- CDROM drive
- Soundcard: CraetiveLabs® Audigy® or Audigy2®
- Operation system: Microsoft® Windows98®, Millennium®, XP®

The configuration above is comfortably suitable for everyday usage of the program.

Advantages of using PC and soundcard versus standalone test equipment:

- PC can be used for other purpose; it does not need to be bought for only this project.
- Easy and fast software development.
- Complex measurement function can be carried out.

- Everything is at your hand, does not require additional electronics only an optional switch
- Very nice instruments and graphic display can be implemented.
- Easy software upgrade and distribution.
- It does not require special knowledge.

Disadvantages of using PC and soundcard:
- Mysterious problems of operating system, programs and hardware.
- PC is unusual equipment on the bench top, it requires extra space and sometimes uncomfortable to use it.
- If it is not available, it is very expensive to buy only for this purpose.

The lots of advantage suggest that they have to be exploited and should be applied for balancing.

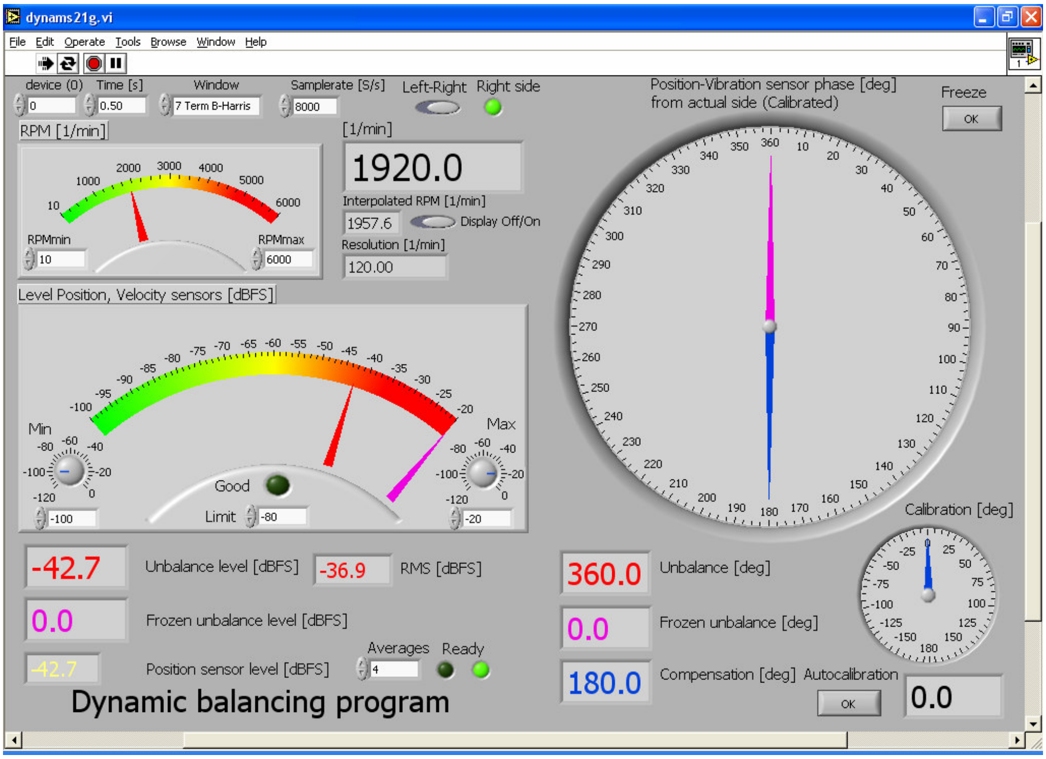


**Figure 15 Software screen**

In the software the following functions were implemented:
- It makes a Fourie transformation for the two channels by FFT. All FFT parameters can be set.
- It selects the highest amplitude component from the position reference channel spectrum and calculates the revolution speed and sensor input level from it.
- The interesting RPM range can be set and the peak amplitude is searched only in this range.
- It measures the amplitude of the vibration sensor and phase between position reference and vibration channel at the revolution frequency as described above. It is a real selective voltage and phase measurement, what is very sensitive and remains exact at low level also.
- It measures the RMS (full band without filter, total power measurement) level of the vibration sensor. The noise and other harmonic components are included this

measurement. The all signal and the first order signal can be compare in order to make a signal-noise ratio measurement.

- It selects the right and left side by external optional switch.
- It displays levels of the sensors, phase between position reference sensor and vibration sensor (unbalance phase) and opposite of this phase (compensation phase, where extra weight should be placed) and certainly does it for every two sides.
- The phase should be always seen from actual side. The right channel is normal phase and the left is inverted (mirrored). The clockwise is positive.
- It can calibrate (zeroing) the phase for two sides.
- It can take moving average for level of every two sides. It sums the last n measurement and divide the sum by n. It allows more precise level measurement. Ready green "LED" lights to show average finished.
- It can freeze the last level and phase measurement for every two sides, in order to ease the balancing result comparison with the last measurement.
- It has a level limit for the end of balancing. When the unbalance is lower than a predetermined limit, the green "LED" lights to show the good result of the balancing.
- If the position reference signal has more or less pure sine wave and really larger revolution frequency component than other, then the program can interpolate the revolution speed. This method has better resolution than the resolution of FFT bins, but requires additional processor power. Therefore this function can be switched off.
- When quitting from the program all parameters are saved, and at the next program starting it will be recalled.

This list suggests that it is so complicated task to use dynamic balancing program, but it is not true. The usage of the program is very simple and it can be learnt very quickly. The advantages are worth the time, but the following fact should be kept in your mind.

Facts about using this signal processing method:

- Selective amplitude measurement sensitive only for the unbalance caused component of the whole signal versus the instrument using peak or RMS detector. (Please, compare RMS reading with unbalance level display, the RMS value contains noise, harmonics and blade frequency components).
- If you want correct amplitude measurement, use Flat-top or 7-term Blackman-Harris window for the sample weighting. The first is the best.
- Using longer acquisition time can reduce the phase and the amplitude uncertainty of the measurement and smaller amplitude can be measured by this way. The phase measurement can be made at lower level of unbalance.
- Using longer acquisition time causes finer resolution of the revolution speed ($\Delta f=1/T$, $\Delta RPM=60\Delta f$). It means the RPM reading will be more exact.
- The maximum balancing RPM depends on the samplerate. $RPMmax<fs*60/2.5$, it comes from Shanon theory and the analogue low pass filter characteristics, but if higher samplerate is used with the same frequency resolution, then it requires more computational time. Generally 8000S/s is recommended.

| Samplerate [S/s] | RPMmax [1/min] |
|---|---|
| 8000 | 192000 |
| 11025 | 264600 |
| 22050 | 529200 |
| 41100 | 1058400 |

**Table 1 Sample rate vs. maximum allowable revolution speed**

In the table only the first harmonic is taken into account.

- The program displays the input voltage of the sensors in dBFS, it is a logarithmic unit referred to full scale input voltage of the soundcard analogue-digital converter. It means dBFS= 20log ($U/U_{FS}$). This unit makes our life simpler, don't have to remember for small numbers. In general dB means amplitude ratio, for example 20dB decrease in amplitude equal amplitude decreased to 1/10 of original amplitude.

| dB | Voltage, velocity ratio |
|---|---|
| 1 | 1.12 |
| 2 | 1.25 |
| 3 | 1.41 |
| 5 | 1.77 |
| 6 | 2 |
| 7 | 2.23 |
| 8 | 2.5 |
| 9 | 2.81 |
| 10 | 3.16 |
| 20 | 10 |
| 30 | 31.6 |
| 40 | 100 |
| 50 | 316 |
| 60 | 1000 |

**Table 2 dB vs. voltage ratio**

The measured voltage is proportional with the vibration velocity (in the case of using balancing machine described above), but the

$$\mathrm{F} = \mathrm{Ma(t)} = \mathrm{mr}\omega^2 \sin(\omega t)$$

**Equation 3 Rotating unbalance caused acceleration in the case of ideal soft suspension (no suspension force)**

(M weight of the rotor or the whole turbine (it depends on only the rotor or the whole turbine is balanced), m unbalances weight), this means that acceleration proportional with the unbalance ($mr$). The velocity is the integral of the acceleration.

$$v = \int_0^{\tau} \frac{mr}{M} \omega^2 \sin(\omega t) dt + v_0 = -\frac{mr\omega}{M} \cos(\omega t) + v_0 \quad \textbf{(In our case } \mathbf{v_0=0)}$$

**Equation 4 Acceleration vs. velocity for the sinusoidal vibration**

It means that unbalance reading increases 20dB/decade if RPM increases and it has 90 degree phase shift relative to unbalance position (The equations above are true only in the idealistic case).

When the turbine is in its own place and in the general case there is no handy recognisable general relation between rotor unbalance and the machine vibrations. The unbalance response depends essentially on speed, the geometric proportions and mass distribution of rotor, as well as on the dynamic stiffness of the shaft, bearings and the foundation. Machine stiffness is unknown to owners in most cases. Moreover, combining all of these factors will truly result in complicated equations between the unbalance and resulting vibration. In other words, for a particular rotor, unbalance vibration will have different values depending on its operating speed, type of bearings (e.g. fluid film or rolling element), foundation etc. while the unbalance amount itself is constant and only related to the rotor. So, the balancing quality limit should not be oversimplified and given through vibration readings only. This is especially true for new machines for which no pervious vibration experience exists[3]. ISO 1940/1 concerns to rigid rotor only not the whole machine. This standard provides generalised grades for which rotor application, mass and speed. High-speed model gas turbine is out of the speed range of the strictest G0.4 grade.

- The reference position signal should contain strong revolution frequency signal, in order that the program can identify the revolution frequency. This means that revolution frequency component has to be the largest amplitude component of the signal. The signal level should be in the interval of –20 and –3dBFS. Changing the distance between the sensor and magnet can control the level of the signal.
- Peak value of reference position signal and vibration signal should be lower than 2Vp-p at the input of the soundcard.
- The instrumentation and operator error limits the residual unbalance the exact value is very elaborate to determine. The residual unbalance can be determined by putting a calibration weight to every 45-degree position step by step. The value of unbalance should be determined for every eight position separately. The mean value of unbalance level comes from the calibration weight and the maximum deviation from mean value represents the residual unbalance.

This software can be used for identifying shaft bending with two accelerometers on the house of the bearings under working conditions (running engine). The accelerometers should be mounted opposite directions and axially on the bearing house and if there is shaft bending then the revolution frequency (first order) component of the signal of the accelerometer are in phase (0 degree). This method is not scope of this article currently.

## THE PRACTICE

At first some idea will be described about system check:

**1.** The sensors level check: When shaft rotates, the reference signal level should be in the interval of –20 and –3dBFS. Changing the distance between the sensor and magnet can control the level of the signal. The left and right side sensor level depends on the unbalance, if put a relatively heavy weight to the proper wheel, it has to be around –40÷-10dBFS. **Keep your eyes on the reference position signal! It is the base of the measurement.**

[3] 11. AL-SHURAFA, Ali M., Determination of Balancing Quality Limits, Vibration Engineer, Saudi Electricity Company- Ghazlan Power Plant, Saudi Arabia

2. Phase stability check: Put a relatively heavy weight (plasticine) to only one side of the rotor. Control the revolution by the compressed air flow rate (valve or regulator) between 2000-6000RPM (2000-4500RPM) range very slowly and measure the phase (acquisition time 0.5s, sample rate 8000 1/s, average 1). The phase variation should be less than $\pm 10 \div 20°$. If not, should check the sensor being squeezed or there is looseness in the mechanics. The good phase stability is never enough but it is a necessary condition, the constant phase - as the experience shows - can be a sign of completely getting stuck or deteriorated sensor system, but it measures the amplitude of the vibration. This check should be separately done for two ends of the turbine.
3. Phase direction and functional check: Choose a revolution (RPM) with avoiding mains frequencies (50 or 60Hz corresponds 3000RPM or 3600RPM), 4000RPM (acquisition time 1s, sample rate 8000 1/s, average 4) are good for everybody. Write down the phase and amplitude to be measured without additional weight. Sign a point of the wheel with a permanent pen at the same direction of the two sides. Put a heavy unbalance weight (plasticine) to only one wheel of the turbine at the marker line. The amplitude and phase should be different from original readings. Calibrate the phase and after move the weight with 90 degrees clockwise. The readings should show the real value. It should be done for 180, 270 degree and separately for other side. If every two side show opposite phase, the wires of the revolution sensors should be inverted. If only phase of the one side is inverted, then only wires of this side should be inverted. **This test with above one shows that everything is working well and enough to do it as daily routine before using balancing system.**
4. Mechanical resonance frequency check: **It should be done only one time after assembling of the balancing machine.** Put a heavy unbalance weight (plasticine) to only one wheel of the turbine and slowly increase the revolution from 1000 to 6000 RPM (acquisition time 0.5s, sample rate 8000 1/s, and average 1). Measure the phase and amplitude of the given side. Around 1400RPM (23Hz) the phase and amplitude should be change very sharply, but this resonance point depends on the weight of turbine and the suspension. It must be lower the 2000RPM. The total phase changing is about 180 degrees when increasing or decreasing the speed of the shaft. The resonance displacement is bigger than in normal case, but the velocity sensor has its lower pass band frequency limit here, therefore it is not sure that it shows the amplitude deviation. This resonance frequency should be kept lower than rotational frequency used for balancing. It is can be done with decrease of spring force of the suspension (soft suspension).
5. Cross-effect test: Choose a revolution (RPM) with avoiding mains frequencies (50 or 60Hz corresponds 3000RPM or 3600RPM), 4000RPM (acquisition time 1s, sample rate 8000 1/s, average 4) are good for everybody. Select a pre-balanced rotor. Measure the amplitude and phase of the unbalance and after put a weight to only one side. Watch the effect of the weight to one and the other side. Generally it has opposite phase unbalance at the other side and it may cause same level of unbalance at the other side. **Keep this phenomenon in your mind!**
6. Software test: It requires software sound generator and the soundcard output has to be connected to the input of it or select the wave input.

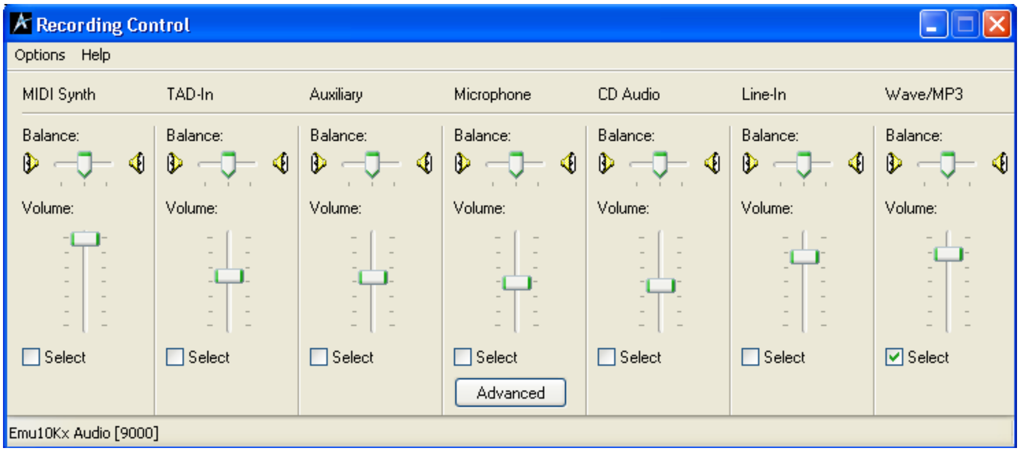


**Figure 16 Mixer configured to wave input in order to test it by software sound generator without any cable**

SgOne from Vic Richardson, DazyWeb Laboratories (dazyweb.labs@verizon.net) is recommended in user mode (rotary switch in U position). The amplitude and phase of the oscillator is freely and separately adjustable side by side.

After everything works well, the balancing can be started. The aim is to decrease the unbalance level of every two sides. The following steps should be done:

1. Choose a revolution (RPM) **with avoiding mains frequencies** (50 or 60Hz corresponds 3000RPM or 3600RPM), 4000RPM (acquisition time 1s, sample rate 8000 1/s, average 4) are good for everybody. Try to keep the revolution always constant by controlling the compressed air valve or pressure regulator to avoid the amplitude uncertainty.
2. Sign a point of the wheel with a permanent pen at the same direction of the two sides. Calibrate the phase by putting bigger weight (at the marked point) than unbalance only to the left side and after press the auto button. This position will be the reference phases (zero degree). The weight place has to be the same plane where the shaft will be ground. Repeat it for right side also (removing weight from left and put to the right). The side can be selected by left/right button, if switch installed, then the selection is automatic, if not it should be done with the connectors. **After the first successful balancing round –when the unbalance successfully decreased- it should be repeated. And time after time it should be done to make the phase reading more exact with the lower value of unbalance.** If it is not done, this error can make phase rotating of unbalance position, the exact unbalance position can not be known. The removed weight will not be at the correct place and the balancing has been never finished. It is caused by not being able to put much heavier weight than the unbalance of the rotor under balancing, therefore the phase calibrating will be more exact when the unbalance decreasing. Besides the disadvantages of phase calibrating, currently it is the simplest method to eliminate the phase error of mechanics and sensors.

**Figure 17 János Horváth is grinding his turbine**

3. Balancing of the heavier wheel has to be started, generally it is the turbine (left side). Where the phase shows the unbalance, there is where the grinder has to be used. Continue the balancing until the other side getting better (unbalance decreasing) and the corrected side also getting better. After this point (when the other side getting worse) go to the other side (right side) and repeat the process (iterating) until reaching the requested balancing grade (go back to the first side and vice versa). Generally it is the noise level, around -80÷-100dBFS (Avoid the vibration and shock of the carrying bench). This level can be determined by removing sensors and replaced by a short wire. 50-60dB decreasing in unbalance level is right enough. Other sign of reaching limit of the method, when the phase is getting unstable (the phase instrument needle rotating around). Setting longer acquisition time provides lower established balancing grade, but it is uncomfortable. Sometimes higher effective sensitivity can be achieved by higher gain setting at the mixer of the soundcard.
4. The sign of doing it well is the decreasing vibration level at the every two sides.
5. Unfortunately the cross-effect makes the things worse; therefore sometimes the balancing is an intuitive process and requires lots of experience.

6. After the first run of the engine, recommended repeating the balancing of the engine, because the small movement of rotating parts.

## CONCLUSION

The program and mechanics proved to be useable for balancing Phoenix Mk-4. There are some disadvantages of this balancing solution, but these can be corrected by enhancing the program and rebuilding the mechanics. The future plans for making better balancing systems are as follows:

- Trying to make a universal balancing bench without traverse bearings (for any kind of small turbine). The turbine will be fixed with iron prism and belts.
- Implementing an automatic phase stability check and phase averaging.
- Cross-effect eliminating with calibration.
- Putting into the practise the trial weight balancing method to eliminate the phase calibrating problems and this method can suggest the quantity of having to remove weight.
- Working out the revolution speed in range unbalance measurement.
- Realising the balancing unit (gmm) calibrating possibilities.
- Developing a propeller-balancing machine.
- Measuring the vibration of turbine and using the data for diagnostic purpose.

I hope that this article calls the turbine builder's attentions to this practical approach of the balancing solution and can help other people to avoid pitfalls and to solve their problems. Many thank to my wife for her patient and to János Horváth for his continuous support. If you have any questions related to this article, please, do not hesitate to contact me.

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