"Super Spring"—A Long Period Vibration Isolator

R. L. Rinker and J. E. Faller*

Joint Institute for Laboratory Astrophysics, National Bureau of Standards and University of Colorado, Boulder, CO 80309

We have devised a new mechanical isolating device which we call a "super spring." The super spring isolator makes use of the fact that a mass suspended by a long spring is effectively isolated (from vibrations) for all frequencies higher than the system's natural resonance. We have developed a method of electronically terminating a 30 cm-long spring in such a way that the mass suspended from it behaves as if the spring were one kilometer or longer in length. This permits us to provide isolation for frequencies as low as 0.02 Hz. We will discuss the principle, the results of shake-table tests, and the implications of this technique for measurement science.

Key words: inertial reference; long period seismometer; mechanical vibration isolation.

1. Introduction

This preliminary determination I generally make the night before the deflections and periods are determined, which in Oxford is best done on Sunday night between midnight and 6 a.m. The daytime, of course, is out of the question, owing to the rattling traffic on the streets in St. Giles', about a quarter of a mile away; and all nights except Sunday night the railway people are engaged making up trains and shunting, which is more continuous and disturbing to the steadiness of the ground than a passing train. Even these come through at intervals on a Sunday night, and this limits the accuracy with which the periods can be observed.

The traffic and the trains are not the only causes of disturbance. Wind, by pressing upon the building and neighboring trees, of course shakes the ground; but on Sept. 9–10, a particularly quiet night, I had to leave, owing to a sudden disturbance producing a pendular motion of 15 divisions, or 150 units, and for some time there was not quiet. As the motion was clearly produced by a lurch of the whole instrument and table carrying it, and was greater in amount than any traffic in the busiest part of the day had ever produced, and was moreover free from the high period tremor characteristic of human disturbance, I at once set it down to an earthquake... I had hoped to have made a greater number of experiments under more widely differing conditions, but the strain which they entail is too severe, for not only have I had to give up holidays for the last three years, but to leave London on Saturdays and occasionally to sit up all Saturday and Sunday nights at the end of a week's work. The conditions, therefore, are too difficult for such an extended series as I should like to make to be possible, and I must after one or more effort, leave the problem to others who have leisure, and what is of far greater consequence, a quiet country place undisturbed by road and railway traffic, and who possess the knowledge and manipulative skill which the experiment requires...

Professor C. V. Boys "On the Newtonian Constant of Gravitation" Philosophical Transactions of the Royal Society of London for the Year MDCCXXCV. Volume 186.

*Staff Member, Quantum Physics Division, National Bureau of Standards.

Isolation of experiments from mechanical vibrations has been a long-standing problem for experimental physicists. Since the time of C. V. Boys, experiments have become increasingly more sophisticated and sensitive and this has served only to increase the problems resulting from vibration. And while measurements have increased in accuracy and complexity, the approaches to vibration isolation have remained at a rather simple level. Today, commercial isolation systems are available which provide isolation from frequencies greater than 2 Hz. However, for experiments sensitive to frequencies of 1 Hz and below, these systems are not helpful. In particular, today's state-of-the-art absolute gravity measurements require isolation at frequencies much lower than 1 Hz. Further, the low frequency limit of earth-bound gravity wave experiments will be limited by the character and the quality of the isolation that can be afforded. While at first glance it might seem that the problem of lowering the isolation regime ("corner" frequency of an isolator) is a simple matter of getting weaker springs and larger masses, there are very real practical limitations. A simple calculation shows that the resonant frequency of a mass $m$ which stretches (by a distance $x$) a spring of spring constant $k$ is equal to the frequency of a simple pendulum whose length $l$ is equal to the amount of this stretch:

$$mg = kx = kl$$

$$\omega = \left[ \frac{k}{m} \right]^{1/2} = \left[ \frac{g}{l} \right]^{1/2}$$

Thus a spring with a period of 60 sec would have to be stretched about 1 km—which rules out its use for normal laboratory applications. Since the frequency of a mass-spring system depends on $g$ and the stretch length of the spring, it is clear that in order to achieve low-frequency isolation one must do something other than attempt to straightforwardly utilize a simple spring system.

To overcome these practical limitations, we have developed what we call a "super spring"—a mass supported by a finite length spring but whose period—and isolation characteristics—is that of a spring very much longer. In order to understand how one might get a simple spring system of tractable size to behave as if it
were, say 1 km in length, imagine a mass hanging from a spring stretched 1 km. As the mass oscillates up and down all the coils of the spring oscillate with the same period. The coils near the bottom would have an amplitude nearly equal to the amplitude of the mass, while the coils near the top would have an amplitude that is small compared to the amplitude of the mass. At the half-way point, the amplitude would be exactly half. If we were to study the motion of a coil 30 cm above the mass as the rest of the spring was lengthened, we would find that the longer the spring got, the closer this coil would track the mass. In fact, if the spring were infinite in length, the coil’s motion would track the mass exactly. It is this behavior of spring-mass systems that gave us the clue as to how to terminate the top portion of a spring at 30 cm so as to make the supported mass act as if it were suspended from a spring 1 km in length. If one were to grasp the coil 30 cm above the mass and then somehow continue to move it up and down in exactly the same way it was moving with a km of spring above it, one could then cut off the top km and still have a spring that would act if it were 1 km long. Since this involves causing a particular point on the spring to track the bottom, a servo approach can be used to transform an ordinary length of spring into a super spring.

Figure 1 is a schematic representation of the super spring. The system consists of a bracket supported by two springs which supply the dc force to support the bracket and the attached mass-spring system (or in terms of the previous discussion they supply the force that the balance of the 1 km of spring would have supplied). From this bracket, the main spring and mass are hung. The top of this spring is made to track the motion of its bottom utilizing a position detector which measures the length of the main spring with respect to the bracket. Light from a light emitting diode (LED) is focused by a ball (serving as a rotation insensitive lens) onto a split photodiode. The currents from the two halves of the diode are amplified, by a current controlled voltage source, and the resulting voltages are then differentiated. This supplies an analogue voltage that is a measure of the position of the mass with respect to the bracket. This analogue voltage is passed through a servo compensating amplifier which drives a current through a voice coil centered in a loudspeaker magnet, which results in a force on the bracket. This forces the bracket (and thus the top of the main spring) to track the mass. The degree of tracking is simply related to the gain of the servo system. The higher the gain the better the tracking—i.e., the closer the coils track the motion of the bottom mass—and therefore the longer the system period.

At this point a brief discussion of the isolation characteristics of a simple spring-mass system is in order.

2. Theory

Figure 2 shows two simple spring-mass circuits: (a) is called the absolute damping case, and (b) is the relative damping case. By writing the equations of motion for these systems and solving for \( \frac{X_2(S)}{X_1(S)} \) (the system transfer function) we get for (a)

\[
\frac{X_2(S)}{X_1(S)} = \frac{\omega_0^2}{S^2 + \frac{\beta}{m} S + \omega_0^2}
\]

and for (b)

\[
\frac{X_2(S)}{X_1(S)} = \frac{\omega_0^2}{S^2 + \frac{\beta}{m} S + \omega_0^2}
\]

where \( S \) is the Laplace transform variable (\( \sigma + j \omega \)), \( \beta \) is the system’s damping coefficient, \( k \) is the spring constant, \( m \) is the mass, and \( \omega_0^2 = k/m \). The isolation characteristic is found by evaluating the magnitude of the transfer function for \( S = j \omega \). For (a) we get

\[
\frac{|X_2(j \omega)|}{|X_1(j \omega)|} = \left(\frac{\omega_0^2 - \omega_0^2}{\omega_0^2} + \frac{\beta^2}{m^2} \omega_0^2\right)^{1/2}
\]

and for (b)

\[
\frac{|X_2(j \omega)|}{|X_1(j \omega)|} = \left(\frac{\omega_0^2 + \beta^2}{m^2} \omega_0^2\right)^{1/2}
\]
Since spring-mass systems isolate only for frequencies $\omega$ that are greater than $\omega_0$, the two expressions can be simplified for $\omega > \omega_0$. Examination of these asymptotic forms reveals a basic difference between the two systems. For (a), the "absolute" damping case, we get

$$
\frac{X_2(j \omega)}{X_1(j \omega)} \sim \frac{1}{\omega^2} \quad \omega \gg \omega_0 \ ;
$$

(6)

for (b), the "relative" damping case, we get

$$
\frac{X_2(j \omega)}{X_1(j \omega)} \sim \frac{1}{\omega} \quad \omega \gg \omega_0 \ .
$$

(7)

This difference in the asymptotic behavior leads one to the conclusion that, for an isolation device, the "absolute" damping case is preferable. The problem, however, is that the body of the dash pot has been assumed to be connected to an inertial reference. This being available, however, assumes that we have already solved the problem.

The simple spring-mass circuit that models the super spring is shown in Fig. 3 where $K_1$ is the spring constant of the support springs, $\beta_1$ is the damping coefficient of the support springs, $m_1$ is the mass of the bracket, $K_2$ is the spring constant of the main spring, $\beta_2$ is the damping coefficient of the main spring, $m_2$ is the mass of the test mass, and $f(t)$ is the force supplied by the voice coil and speaker magnet.

![Figure 3. Spring-mass circuit for super spring.](image)

The equations of motion are two coupled second-order linear differential equations:

$$m_1 \ddot{x}_1 = K_1(x_3 - x_1) + \beta_1(\dot{x}_3 - \dot{x}_1) + K_2(x_2 - x_1) + \beta_2(\dot{x}_2 - \dot{x}_1) + f(t) \ ,$$

$$m_2 \ddot{x}_2 = K_2(x_1 - x_2) + \beta_2(\dot{x}_1 - \dot{x}_2) \ .$$

(8)

(9)

The key to the modification of the system’s behavior is contained in the term $f(t)$. The exact form of $f(t)$ is determined by the transfer function of the electronics. For this discussion we will use a simplified form for $f(t)$:

$$f(t) = G(x_2 - x_1) + \gamma_5(\dot{x}_2 - \dot{x}_1) \ .$$

(10)

This is the statement that the servo action is proportional plus derivative. The most convenient method for the analysis of the system is to work in the Laplace transform domain.

Substituting Eq. (10) into Eq. (8) and then taking the Laplace transform of Eqs. (8) and (9) we get

$$m_1 X_1(S) S^2 = (K_1 + \beta_1)(X_3(S) - X_1(S)) + [K_2 + G + (\beta_2 + \beta_5)S] X_2(S) - X_1(S)) ,$$

$$m_2 X_2(S) S^2 = (K_2 + \beta_2) X_3(S) - X_2(S)) ,$$

(11)

(12)

where $X_1(S)$, $X_2(S)$, and $X_3(S)$ are the Laplace transforms of $x_1(t)$, $x_2(t)$, and $x_3(t)$, and $S$ is the complex transform variable $(\sigma + j \omega)$. These two equations can be solved for the transfer function of interest:

$$
\frac{X_3(S)}{X_2(S)} = \frac{\gamma_5 S + \beta_5 (\gamma_5 S + \omega_0^2)}{S^4 + S^3 (\gamma_1 + \gamma_2 + \gamma_3) S^2 + S^2 (\omega_0^2 + \omega_1^2 + \omega_2^2 + \omega_3^2) + S (\gamma_1 \omega_0^2 + \gamma_2 \omega_0^2) + \omega_0^2 \omega_2^2} \ ,
$$

(13)

where $\gamma_1 = \beta_1 m_1$, $\gamma_2 = \beta_2 m_2$, $\gamma_3 = \beta_3 m_1$, $\gamma_5 = \beta_5 m_1$, $\omega_1^2 = K_1/m_1$, $\omega_2^2 = K_2/m_2$, $\omega_3^2 = K_2/m_1$, $\omega_0^2 = G/m_1$. For this particular system we can approximate some of these terms for high servo gain values:

$$\gamma_1 \ll \gamma_5 \quad \omega_2^2 \ll \omega_5^2$$

$$\gamma_2 \ll \gamma_5 \quad \omega_2^2 \ll \omega_5^2$$

$$\gamma_3 \ll \gamma_5 \quad \omega_2^2 \ll \omega_5^2$$

and Eq. (13) becomes

$$
\frac{X_3(S)}{X_2(S)} = \frac{\gamma_5 S + \beta_5 (\gamma_5 S + \omega_0^2)}{S^4 + S^3 \gamma_5 + S^2 \omega_0^2 + S (\gamma_1 \omega_0^2 + \gamma_2 \omega_0^2) + \omega_0^2 \omega_2^2} \ .
$$

(14)

If we factor the quartic in the denominator into two quadratics of the form

$$
\frac{X_3(S)}{X_2(S)} = \frac{(\gamma_5 S + \omega_0^2)^2 (\gamma_5 S + \omega_0^2)}{(S^2 + AS + B)(S^2 + CS + D)} \ ,
$$

(15)

for large values of $\omega_0$ and $\gamma_5$ we get

$$A = \gamma_5 \ , \ B = \omega_0^2 \ , \ C = \frac{\gamma_1 \omega_0^2 + \gamma_2 \omega_0^2}{\omega_5^2} - \frac{\gamma_5 \omega_0^2}{\omega_5^2} \ , \ D = \frac{\omega_0^2 \omega_2^2}{\omega_5^2} \ .$$

The quantity of interest is of course the isolation characteristic $|X_3(j \omega)/X_2(j \omega)|$. For the frequency range $\omega \ll \omega_1/\gamma_1$, $\omega_0/\gamma_5$ we get

$$
\frac{X_3(j \omega)}{X_2(j \omega)} \sim \frac{\omega_0^2 \omega_2^2/\omega_5^2}{[\omega_0^2 \omega_2^2 - \omega^2]^2 + \omega^2 \omega_0^2} \frac{1}{2} \ .
$$

(16)

By comparison to Eq. (4) we see this is exactly the isolation characteristic of a simple spring with resonance $\omega_0 = \omega_0/\omega_0/\omega_0$ and damping constant $C$. This analysis shows that one can in fact lower the "corner" frequency of a mechanical system with feedback, without the dimensions of the apparatus getting out of proportion. There is, however, a difficulty with this technique. Recall that

$$C = \frac{1}{\omega_5^2} \left[ (\gamma_1 \omega_0^2 + \gamma_2 \omega_0^2) - \gamma_5 \omega_0^2 \frac{\omega_0^2}{\omega_5^2} \right] \ .
$$

(17)

Since we started with a high "$Q"$ mechanical system, $\gamma_1$ and $\gamma_2$ are small, and since the servo gain is large the negative term in the expression for $C$ can dominate. When this is the case the damping factor for the low frequency pole in Eq. (15) $[S^2 + CS + (\omega_0^2/\omega_5^2)]$ is negative. This leads to a transient response that is an exponentially growing sinusoid with frequency $\omega_0$, which is clearly not acceptable.

The obvious means of damping the long-period motion is to connect a dash pot from $x_2$ to $x_3$. In the development of super spring, two types of dash pots were tried. The first was a $4 \times 10^{-3}$ cm tungsten rod immersed in a light silicone oil. While this damped the system's motion
quite well, the creep of the fluid up the rod increased the mass of the test mass to a degree that the system simply continued to move down until the rod hit the bottom of the oil container. Remember that even at dc the spring behaves as a very long one (and is therefore very soft): The addition of only a few milligrams of mass to the 500 g mass results in a substantial lowering of the equilibrium position.

To eliminate the creep problem associated with the fluid, a magnetic dash pot was tried. The test mass was placed in a magnetic field with the result that as the mass moved, eddy currents were induced giving rise to damping forces. This method, though quite successful for low servo gain settings, suffered as the servo gain was increased because the interaction of the fringing field of the magnet and the magnetic spring tended to add a negative spring constant, which either canceled or dominated the positive spring constant, making the system unstable.

The damping method we finally decided on modifies the low frequency (ω < 0.628) response of the servo in such a way as to damp the long-period motion. In order to understand how this method works we will consider the simple spring-mass system shown in Fig. 4 where K is the spring constant of the spring, β is the damping coefficient of the system, m is the mass, and f(t) is a time-dependent force which is derived electronically from the displacement (x₁ - x₂).

**Figure 4. Simplified spring-mass circuit for the damped super spring.**

In the Laplace domain the equation of motion can be written as

\[ mS^2X_2(S) = K[X_1(S) - X_2(S)] + \beta S [X_1(S) - X_2(S)] + G(S)[X_1(S) - X_2(S)] , \tag{18} \]

where G(S) is the transfer function of the electronics. Equation (18) can be solved for the transfer function X₂(S)/X₁(S)

\[ \frac{X_2(S)}{X_1(S)} = \frac{\omega^2 + \frac{\beta}{m} S + \frac{1}{m} G(S)}{S^2 + \frac{\beta}{m} S + \omega^2 + \frac{1}{m} G(S)} . \tag{19} \]

From this transfer function we can see that the time domain behavior of the system depends on the mechanical parameters β and ω and also on the transfer function of the electronics G(S). If we choose

\[ G(S) = \gamma S , \tag{20} \]

which is equivalent to saying the force is proportional to the time derivatives of (x₁ - x₂), then Eq. (19) becomes

\[ \frac{X_2(S)}{X_1(S)} = \frac{\omega^2 + \frac{(\beta + \gamma)}{m} S}{S^2 + \frac{\beta + \gamma}{m} S + \omega^2} . \tag{21} \]

This is easily identified as the transfer function of a single spring with a variable damping coefficient (γ + β). If γ < 0, the "Q" of the system is increased. If γ > 0, the "Q" is decreased. The case γ > 0 is an example of using electronic feedback to damp a mechanical system.

While the above analysis demonstrates the concept of electronic damping, there is a practical limitation that is not taken into account in Eq. (20). The limitation comes in the fact that we are not free to choose G(S) in a totally arbitrary manner. The constraints on our choice of G(S) stem from the fact that G(S) is the transfer function of an electronic system, which must be realized using amplifiers having a finite bandwidth. We can, at best, achieve a transfer function G(S) that acts as a differentiator for a restricted frequency range. The optimum G(S) that can be realized is the transfer function of a second-order band pass filter

\[ G(S) = \frac{\gamma S}{S^2 + AS + \omega^2} . \tag{22} \]

This is simply the transfer function of a perfect differentiator with two poles added. One pole must be added for reasons of op-amp stability, the second pole is added to reduce high frequency noise. Using this G(S) in Eq. (19) we get

\[ \frac{X_2(S)}{X_1(S)} = \frac{\left[ S^2 + AS + \omega^2 \right] \left[ \frac{\beta}{m} S + \omega^2 \right]}{\left[ S^2 + \frac{\beta}{m} S + \omega^2 \right] \left[ S^2 + AS + \omega^2 \right] + \frac{\gamma}{m} S} . \tag{23} \]

To understand how the response of this system is changed as we increase the value of γ, we utilized root locus analysis. Figure 5 is a rough sketch of the root locus of the transfer function Eq. (23) as a function of γ. As γ is increased we see that the mechanical poles move toward the σ axis indicating a decrease in the "Q" of

**Figure 5. Sketch of the root locus of the transfer function, Eq. (23), as a function of γ.**
this resonance. But as γ is increased, the electronic poles move toward the \( j \omega \) axis, which indicates an increase in the "Q" of the electronic poles. If γ is increased enough, the electronic poles will go into the right half plane, at which point the system becomes unstable. While these poles are electronic in nature, they will nevertheless cause the mechanical system to oscillate. The total effective damping one can achieve is determined by the trade-off between the increase in the "Q" of the electronic poles and the decrease of the "Q" of the mechanical poles.

In the derivation of Eq. (13) we used for the feedback force \( f(t) \)

\[
f(t) = G(x_2 - x_1) + \gamma_S(\dot{x}_2 - \dot{x}_1).
\]

(24)

This is equivalent to saying the transfer function of the servo is

\[
G(S) = (G + \gamma_S S).
\]

(25)

This of course cannot be the transfer function of any real system, since all real systems have finite gain and bandwidth. When we assume a general form for the transfer function \( G(S) \) of the electronics, then Eq. (13) becomes

\[
\frac{X_2(S)}{X_1(S)} = \frac{(\gamma_S S + \omega_0^2)}{(S + \gamma_S S + \omega_0^2)}.
\]

(26)

For the servo transfer function used in the super spring, which takes into account the finite bandwidth of the servo amplifiers and the low frequency differentiator (used to damp the long period motion), \( G(S) \) has the form

\[
G(S) = \frac{\omega_0^2}{S^2 + 2\eta_0 \omega_0 S + \omega_0^2} + \rho \frac{2\eta_0 \omega_0 S}{S^2 + 2\eta_0 \omega_0 S + \omega_0^2}.
\]

(27)

Here \( \omega_0, \eta_0 \) are the band width and damping ratio for the servo amp, \( \alpha \) is the gain of the second order bandpass filter used as the differentiator to damp the servo, and \( \omega_0, \eta_0, \) and \( \rho \) are the break point, damping ratio, and gain of the second-order bandpass filter used as the low frequency differentiator which damps the long period motion.

The complete analysis of Eq. (26) with this \( G(S) \) involves finding the roots of, and eighth-order polynomial for, various values of \( G, \alpha, \rho, \eta_0, \omega_0, \eta_L \) and \( \omega_L \). This analysis has been done using a digital computer and a FORTRAN program. The results were in no way profound, so they will not be discussed at length. It is sufficient to say that the optimum values for \( \omega_0, \omega_L, \alpha, \) and \( \rho \) were found to depend on the value of \( G \). This value of course is determined by the desired period of the "long spring." Since we wanted this to be an easily variable parameter the electronics was designed so that \( \omega_0, \omega_L, \alpha, \) and \( \rho \) were also easily varied.

3. Mechanical

There are three basic parts to the mechanical system; the bracket assembly, the flexures which constrain the bracket to have only a vertical degree of freedom, and the main and support springs.

The design requirements on the bracket are that its mass must be kept to a minimum while its stiffness be as high as possible. The low mass requirement results from the fact that the "gain" in Eq. (23) of the electronic transfer function \( G(S) \) is divided by the mass of the bracket \( m \). Stiffness minimizes the tendency at high servo gain of exciting the mechanical modes of the bracket. One can also address this problem by applying damping material (e.g., Dux Seal) to lower the "Q" of these modes and thus making them harder to excite. The bracket (see Fig. 6) consists basically of a long tube which is centered in the external housing by means of a flexure at each end. These flexures constrain the bracket to move in only the vertical direction. The housing containing the position detection system, which consists of a light emitting diode and a split photodiode, is attached to the bottom of the tube. The voice coil is attached at the bottom of this housing. The main spring hangs down through the center of the tube which results in the test mass being centered in the position detection housing. The weight of this assembly (tube, housing, voice coil, main spring, and test mass) is carried by the three auxiliary support springs. The voice coil fits into the gap of the speaker magnet—the latter being attached to the external housing. The external housing rests on three leveling screws.
The flexures are one of the most critical parts of the mechanical system. They must be soft to vertical motion, still to sideways motion, and precise enough so as not to cause the tube to tilt as it moves up and down. Several flexure designs were tried. Figure 7 shows the design that ultimately worked the best. They are first machined from $17.5 \times 10^{-3}$ cm thick, 10.8 cm diameter disks of beryllium copper, and then heat treated. These flexures give about 2 arcseconds of tilt for a bracket motion of 0.6 cm.

This signal, processed by a servo-compensating amplifier, drives a loudspeaker voice coil which in turn supplies the needed force to cause it to (nearly) track the motion of the test mass. With an LED current of 200 mA the vertical position sensitivity of this detector is 0.7 V $\mu$m$^{-1}$. The noise level in a 5.0 kHz bandwidth is 5 mV P-P while the noise level in a 1 Hz bandwidth is about 0.5 mV P-P.

4. Test Results

Once the construction of super spring was completed we tested its isolation characteristics. To do this, we constructed a “shake table” on which we could place the spring. This table could produce an amplitude of $25 \times 10^{-3}$ cm at frequencies of less than 5 Hz, and about $2 \times 10^{-3}$ cm at 25 Hz. The table surface, driven by a system of levers and a speaker magnet-voice coil system, is constrained so as to tilt less than one arcsecond—the accomplishment of which proved to be a nontrivial task. The power amplifier used to drive the coil could deliver $\pm 12$ A into the coil. An LED photodiode position detector was used to monitor the table motion. The isolation measurements were made using a HP 3582A Spectrum Analyzer. The noise source from the 3582A was used to drive the table. The output from the table's position detector and the position of the test mass with respect to the floor (“inertial space”) were applied to the two inputs of the spectrum analyzer which computed the transfer function. Figure 8 is an example of such a transfer function. This particular one was taken with the spring period set at 12 s (lower than usual) so that the 1/12 Hz resonance could be seen.

![Isolation transfer function](image)

FIGURE 8. Transfer function obtained during “shake table” testing of super spring.

Further evidence that the spring indeed isolates is seen when the test mass is used to hold the reference cube corner in a free falling absolute gravimeter (Zumberge et al., in these proceedings). Figure 9 shows two histograms of 150 g measurements (one set with the inertial reference and one without) which demonstrate a reduction in scatter by a factor of 20.
5. Conclusions

Toward the end of the same article quoted in part at the beginning of this paper, C. V. Boys states: “I would strongly urge that in such a case, a room more uniform in temperature than the one at Oxford should be employed... I do not think any ready-made room is likely to be found available. A disused adit, at a great distance from existing mining operations, would be perfect.” The use of the super spring concept as we have applied it to isolate in the vertical, or as it could be adapted to isolate in other directions, offers an alternative to Boys' recommendation. Clearly the ideal situation would be to both isolate and operate at a quiet site. In many cases this simply is not practicable. However, in all cases the principles of the super spring can still be used to effectively get new labs for old.

Figure 9. Histogram of measurements of the gravitational acceleration $g$ without and with a super spring isolated reference corner cube.