

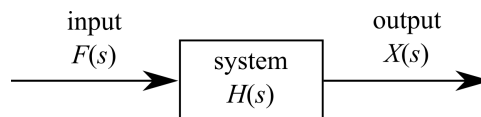
4 Transfer Function for Vibrating Systems

Thusfar, this text has only considered forced vibrations for 1-DOF systems excited with forcing functions that can be easily expressed using either sin or cos examples. Therefore, the previously developed solutions are only acceptable for systems with known and simple excitations. This chapter will introduce the concept of transfer functions for solving vibration related problems. The transfer function, in particular the Laplace transfer function, is an important tool in the study of vibrations as it allows the practitioner to solve for the temporal response of a system for a variety of inputs using a single approach. Examples of force excitation that can be calculated include using this method include:

- sinusoidal
- base excitation
- impulse
- arbitrary input
- arbitrary periodic input

4.1 Transfer Function Method (Generic)

Consider the following system



where $F(s)$ is the input, $H(s)$ is the system, and $X(s)$ is the output from the system. This formulation is called the transfer-function approach and is commonly used for the formulation and solution of dynamic problems in the control literature. It can also be used for solving the various forced-vibration problems including those from complex or stochastic inputs.

4.2 Review of Laplace Transforms

An integral transform is the procedure of integrating the time dependence of a function into becoming a function of an alternative variable, or parameter, which can be manipulated algebraically.

Of interest to this class is the Laplace transform ($\mathcal{L}[f(t)]$) of the function $f(t)$. Here, a Laplace transform is used as a method of solving the differential equations of motion by reducing the computation needed to that of integration and algebraic manipulation.

The definition of the Laplace transform of the function $f(t)$ is:

$$\mathcal{L}[f(t)] = F(s) = \int_0^{\infty} f(t)e^{-st} dt \quad (1)$$

where $f(t) = 0$ for all values of t less than 0. Here, the s is a complex value. Lastly, the term $F(s)$ is a generic term that represents the input to a system. As this class needs the derivative of functions, we will calculate that next:

$$\mathcal{L}[\dot{f}(t)] = \int_0^{\infty} \dot{f}(t)e^{-st} dt = \int_0^{\infty} e^{-st} \frac{d[f(t)]}{dt} dt \quad (2)$$

integration by parts yields

$$\mathcal{L}[\dot{f}(t)] = e^{-st} f(t) \Big|_0^{\infty} + s \int_0^{\infty} e^{-st} f(t) dt \quad (3)$$

Astutely, it can be noticed that the second term $s \int_0^{\infty} e^{-st} f(t) dt$ is the input to the system $F(s)$. Therefor, with a little rearranging this becomes:

$$\mathcal{L}[\dot{f}(t)] = sF(s) - f(0) \quad (4)$$

Successive iteration yields:

$$\mathcal{L}[\ddot{f}(t)] = s^2 F(s) - sf(0) - \dot{f}(0) \quad (5)$$

A few key points of the Laplace transforms are:

- the domain of the problem from the real number line (t) to the complex plane (s).
- The integration of the Laplace transform changes differentiation into multiplication.
- The transform procedure is linear. Therefore, the transform of the linear combination of two transforms is the same as the linear transformation of these functions.
- to move from the time domain to the complex number plane we typically use tables of pre-solved integral.

- The function $x(t)$ can be obtained by taking the inverse Laplace transform defined as $x(t) = \mathcal{L}[X(s)]^{-1}$

The Laplace transform can be calculated in symbolic form. In particular interest to this class is the Laplace form of the system output $X(s)$

$$\mathcal{L}[x(t)] = X(s) \quad (6)$$

$$\mathcal{L}[\dot{x}(t)] = sX(s) - x(0) \quad (7)$$

$$\mathcal{L}[\ddot{x}(t)] = s^2X(s) - sx(0) - \dot{x}(0) \quad (8)$$

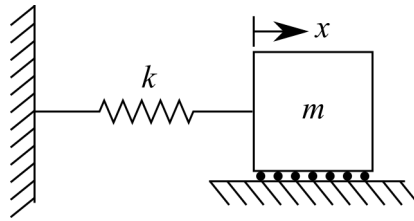
here, $x(0)$ and $\dot{x}(0)$ are the initial values of the function $x(t)$.

The procedure for using the Laplace transform to solve equations of motion expressed as an inhomogeneous ordinary differential equation is:

1. Take the Laplace transform of both sides of the EOM while treating the time derivatives symbolically.
2. Solve for $X(s)$ in the obtained equation.
3. Apply the inverse transform $x(t) = \mathcal{L}[X(s)]^{-1}$

4.3 Transfer Function Method for Free Vibration Systems

Consider the undamped single-DOF system:



The EOM for this system is a homogeneous differential equation because the right-hand side is equal to zero:

$$m\ddot{x}(t) + kx(t) = 0 \quad (9)$$

Here we will leave the “(t)” for clarity to differentiate the time domain solution from Laplace solution “(s)” in the complex plane, as discussed in section 4.2. The EOM can be rewritten in standard form as:

$$\ddot{x}(t) + \omega_n^2 x(t) = 0 \quad (10)$$

where the initial conditions at $t = 0$ are $x(0) = x_0$ and $\dot{x}(0) = v_0$. Taking the Laplace transforms of both sides of the EOM yields:

$$[s^2X(s) - sx_0 - v_0] + [\omega_n^2X(s)] = 0 \quad (11)$$

using equations 6 and 8 from section 4.2. Solving for the output of the system $X(s)$ yields:

$$X(s) = \frac{sx_0 + v_0}{s^2 + \omega_n^2} \quad (12)$$

We can expand this form of $X(s)$ to obtain equations listed in our Laplace Transform table:

$$X(s) = \frac{sx_0}{s^2 + \omega_n^2} + \frac{v_0}{s^2 + \omega_n^2} \cdot \frac{\omega_n}{\omega_n} \quad (13)$$

This becomes:

$$X(s) = x_0 \frac{s}{s^2 + \omega_n^2} + \left(\frac{v_0}{\omega_n} \right) \cdot \frac{\omega_n}{s^2 + \omega_n^2}. \quad (14)$$

Next, using the inverse Laplace transform $x(t) = \mathcal{L}[X(s)]^{-1}$ and the two following Laplace transforms (#5 and #6):

$$f(t) \text{ is } \cos(\omega t) \text{ when } F(s) \text{ is } \frac{s}{s^2 + \omega^2} \quad (15)$$

$$f(t) \text{ is } \sin(\omega t) \text{ when } F(s) \text{ is } \frac{\omega}{s^2 + \omega^2} \quad (16)$$

Therefore, we can obtain the solution for the system output $X(s)$ as:

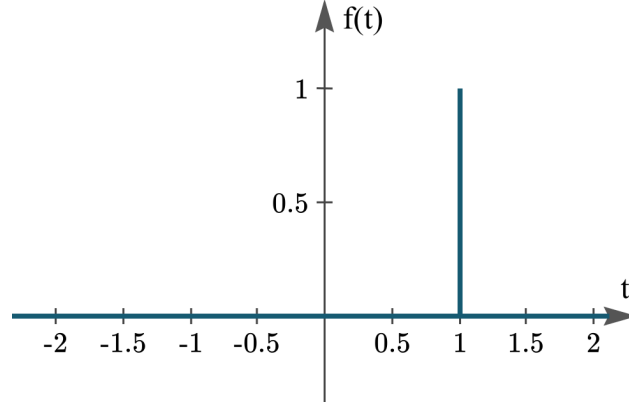
$$x(t) = x_0 \cos(\omega_n t) + \frac{v_0}{\omega_n} \sin(\omega_n t) \quad (17)$$

The same procedure works for calculating the under damped and forced responses, however, in these responses calculating the algebraic solution for $X(s)$ often results in quotients of polynomials in s . These Polynomial ratios may not be found in simple Laplace tables and must be solved using the **method of partial fractions**. An example of this procedure can be found in Appendix B of Inman.

4.3.1 Impulse Response Function

A very common source of vibration is the sudden application for a short-duration force called an impulse. An impulse excitation is a force that is applied for a very short, or infinitesimal, length of time and represents one example of a shock

loading. An impulse is a nonperiodic force that is represented by the symbol δ . The response of a system to an impulse is identical to the free response of the system to certain initial conditions. This is illustrated in the following, in many useful situations the applied force $F(t)$ is impulsive in nature (i.e., acts with large magnitude for a very short period of time).



The impulse response function can be solved for analytically, however, we will solve it using the transfer function approach. Here we will consider the under-damped spring-mass system. First, assume that the system is at rest (no initial conditions). Next, we write the EOM as:

$$m\ddot{x} + c\dot{x} + kx = \delta(t) \quad (18)$$

Taking the Laplace transform of both sides of the equation yields

$$m(s^2X(s) - sx(0) - \dot{x}(0)) + c(sX(s) - x(0)) + kX(s) = 1 \quad (19)$$

However, **if we assume zero initial conditions** (a system at rest when the impulse happens), the equation simplifies to.

$$ms^2X(s) + csX(s) + kX(s) = 1 \quad (20)$$

or

$$(ms^2 + cs + k)X(s) = 1 \quad (21)$$

note that the $\mathcal{L}[\delta] = 1$ per #1 in the transform table. Solving this equation for $X(s)$:

$$X(s) = \frac{1}{m} \cdot \frac{1}{s^2 + 2\zeta\omega_n s + \omega_n^2} \quad (22)$$

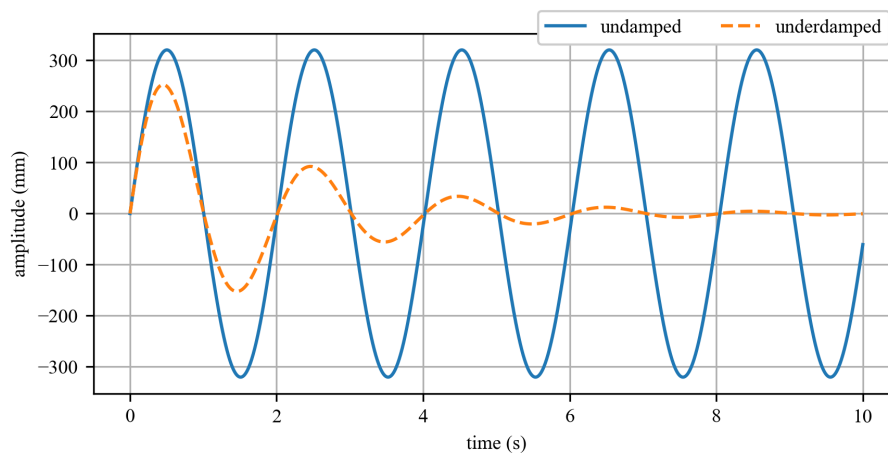
Again, the mass is extracted to develop a formulation that can be found in the Laplace tables. Setting the constraint that $\zeta < 1$ and consulting #10 in the table for Laplace transforms results in:

$$x(t) = \frac{1}{m\omega_d} e^{-\zeta\omega_n t} \sin(\omega_d t) \quad (23)$$

where this is the general solution for a damped system subjected to an impulse loading function. For the undamped case a solution can be obtained by setting $\zeta = 0$. This Results in the following form for the undamped case:

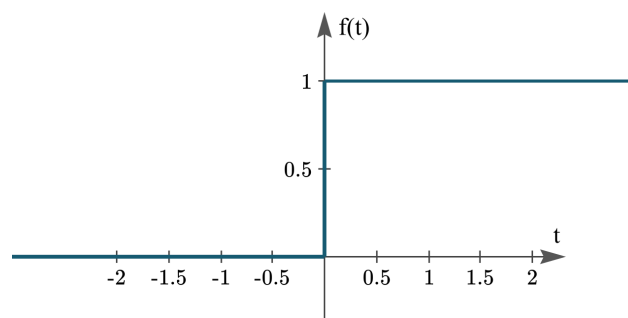
$$x(t) = \frac{1}{m\omega_n} \sin(\omega_n t) \quad (24)$$

Below is a typical response for both a undamped and underdamped 1-DOF system subject to an impulse response at $t = 0$ seconds.



4.3.2 Unit step function

Now consider a unit step function Φ :



A step function is a common loading situation and can represent the dropping of a load into a truck, a car going over a curve, or a motor starting up.

The Laplace transform of the function, for a unit step function Φ , is:

$$\mathcal{L}[\Phi(t)] = \int_0^{\infty} e^{-st} dt = -\frac{e^{-\infty}}{s} + \frac{e^{-0}}{s} = \frac{1}{s}$$

This also lines up with Laplace Transform # 3 from the attached table. This would be expected as Φ is used to represent the unit step function (i.e. a step function with a displacement of 1). As we consider linear systems in this class, we can scale the magnitude of the response by the magnitude of the impulse after the transform is performed.

4.3.3 Undamped spring-mass system

For a spring-mass system subjected to a unit step, **assuming both initial conditions are zero**, the solution can be obtained using the transform method. First, the EOM is

$$m\ddot{x}(t) + kx(t) = \Phi(t) \quad (25)$$

Taking the Laplace transform of both sides and assuming zero initial conditions yields:

$$ms^2X(s) + kX(s) = \frac{1}{s} \quad (26)$$

Next, this equation is solved for $X(s)$ as:

$$X(s) = \frac{1}{s(ms^2 + k)} \quad (27)$$

This can be rearranged as:

$$X(s) = \frac{1}{m} \cdot \frac{1}{s(s^2 + \omega_n^2)} \quad (28)$$

where $\frac{1}{m}$ will pass through the Laplace function. Therefore, taking the inverse Laplace transform yields:

$$x(t) = \frac{1}{m} \cdot \frac{1}{\omega_n^2} (1 - \cos(\omega_n t)) = \frac{1}{k} (1 - \cos(\omega_n t)) \quad (29)$$

4.3.4 Under damped spring-mass system

For a spring-mass-damper system subjected to a unit step, assuming both initial conditions are zero, the solution can be obtained using the transform method. First, the EOM is:

$$m\ddot{x}(t) + c\dot{x} + kx(t) = \Phi(t) \quad (30)$$

Converting to the standard form results in:

$$\ddot{x}(t) + 2\zeta\omega_n\dot{x} + \omega_n^2x(t) = \frac{1}{m} \cdot \Phi(t) \quad (31)$$

taking the Laplace transform of both sides and assuming zero initial conditions yields:

$$s^2X(s) + 2\zeta\omega_nsX(s) + \omega_n^2X(s) = \frac{1}{m} \cdot \frac{1}{s} \quad (32)$$

Next, this equation is solved for $X(s)$ as:

$$X(s) = \frac{1}{s^2 + 2\zeta\omega_ns + \omega_n^2} \cdot \frac{1}{m} \cdot \frac{1}{s} \quad (33)$$

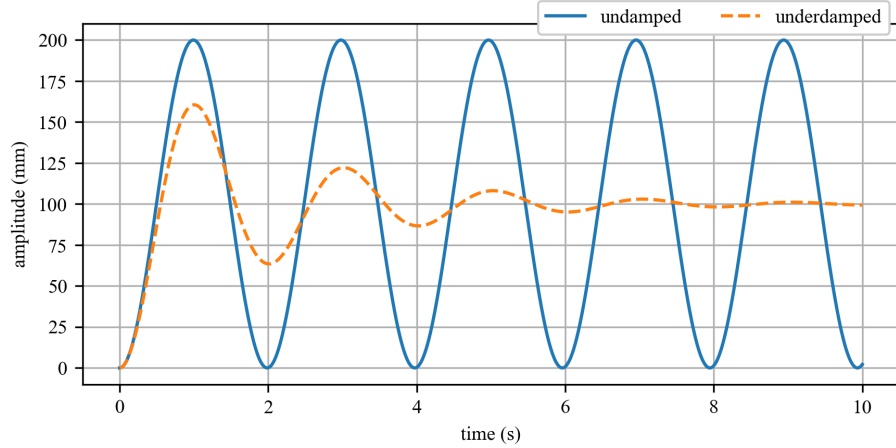
multiplying the right-hand-side of this equation by $\frac{\omega_n^2}{\omega_n^2}$ results in:

$$X(s) = \frac{1}{m\omega_n^2} \cdot \frac{\omega_n^2}{s(s^2 + 2\zeta\omega_ns + \omega_n^2)} \quad (34)$$

Again, the $\frac{1}{m\omega_n^2}$ will pass through the Laplace function. Therefore, taking the inverse Laplace transform yields:

$$x(t) = \frac{1}{m\omega_n^2} \cdot \left(1 - \frac{\omega_n}{\omega_d} e^{-\zeta\omega_nt} \sin(\omega_dt + \phi)\right), \text{ where } \phi = \cos^{-1}(\zeta), \text{ where } \zeta < 1 \quad (35)$$

After obtaining equations for the undamped and under damped cases, the responses for the unit step, solved with the transform method, can be plotted as:



Note that the system will settle out around F_0/k where $F_0\Phi$ is a scaling factor for the step loading.

Example 4.1 A load of roosters m_r is dropped on the floor of a truck bed. Assuming the roosters do not move and that the truck bed is modeled as a spring-mass-damper system (of values k , m , and c , respectively). The load is modeled as a force $F(t) = m_r g$ applied to the spring-mass-damper system, as illustrated in the following figure. This allows a crude analysis of the response of the truck's suspension. First assume that the truck's damper is broken, how does the maximum dynamic displacement compare to the static displacement. What would happen to the maximum displacement if the damper was repaired on the truck?



Figure 1: Dump truck being loaded with roosters showing (a) roosters going into the truck bed; and (b) the single-degree-of-freedom vibration model.

Solution: First, setting the load applied to the truck as 1 unit, it can be seen that this is a unit step loading condition and a broken damper represents an

undamped case. To obtain a rough idea about the nature of static and dynamic displacement, the undamped displacement is

$$x(t) = \frac{1}{k}(1 - \cos(\omega_n t)) = \frac{m_d g}{k}(1 - \cos(\omega_n t)) \quad (36)$$

This equation has a maximum amplitude when the $\cos(\omega_n t) = -1$, resulting in:

$$x(t) = \frac{m_d g}{k}(1 - (-1)) \quad (37)$$

This can be rearranged for the maximum displacement value x_{\max} as:

$$x_{\max} = 2 \frac{m_d g}{k} \quad (38)$$

This is twice the static displacement (i.e., twice the distance the truck would be deflected if the dirt were placed gently and slowly onto it). Thus, if the truck were designed with springs based only on the static load, with no margins of safety, the springs in the truck would potentially break, or permanently deform, when subjected to the same mass applied dynamically (e.g., dropped) into the truck. Hence, it is important to consider the vibration (dynamic) response in designing structures that could experience dynamic loading.

Example 4.2 In testing, an hammer is used to excite a 1-DOF system with an impact (i.e. impulse), however, the hammer ascendantly impacts the system twice. The first impact has a force of 0.2 N, while the second has a force of 0.1 N and happens 0.1 seconds after the first impact. Plot the response for the double impact. The system has the parameters $m = 1$ kg, $c = 0.5$ kg/s, $k = 4$ N/m.

Solution: First, we can define the forcing function as:

$$F(t) = 0.2\delta(t) + 0.1\delta(t - \tau) \quad (39)$$

where τ is the offset between the first and second impacts. Next, considering that the unit impulse has a magnitude of 1 we can obtain solutions for the first impact by first writing it's EOM:

$$m\ddot{x} + c\dot{x} + kx = 0.2\delta(t) \quad (40)$$

Taking the Laplace transform of both sides of the equation yields

$$m(s^2X(s) - sx(0) - \dot{x}(0)) + c(sX(s) - x(0)) + kX(s) = 0.2 \quad (41)$$

However, assuming zero initial conditions, the equation simplifies to.

$$(ms^2 + cs + k)X(s) = 0.2 \quad (42)$$

Solving this equation for $X(s)$:

$$X(s) = \frac{0.2}{m} \cdot \frac{1}{s^2 + 2\zeta\omega_n s + \omega_n^2} \quad (43)$$

Again, consulting #10 in the table for Laplace transforms results in:

$$x_1(t) = \frac{0.2}{m\omega_d} e^{-\zeta\omega_n t} \sin(\omega_d t) \quad (44)$$

where this is the general solution for a damped system subjected to an impulse loading function. The second impact can now be solved for using the same method. However, now the time (t) must be offset by (τ) to allow the impact to still be located at $t = 0$ in terms of the second impact. This results in:

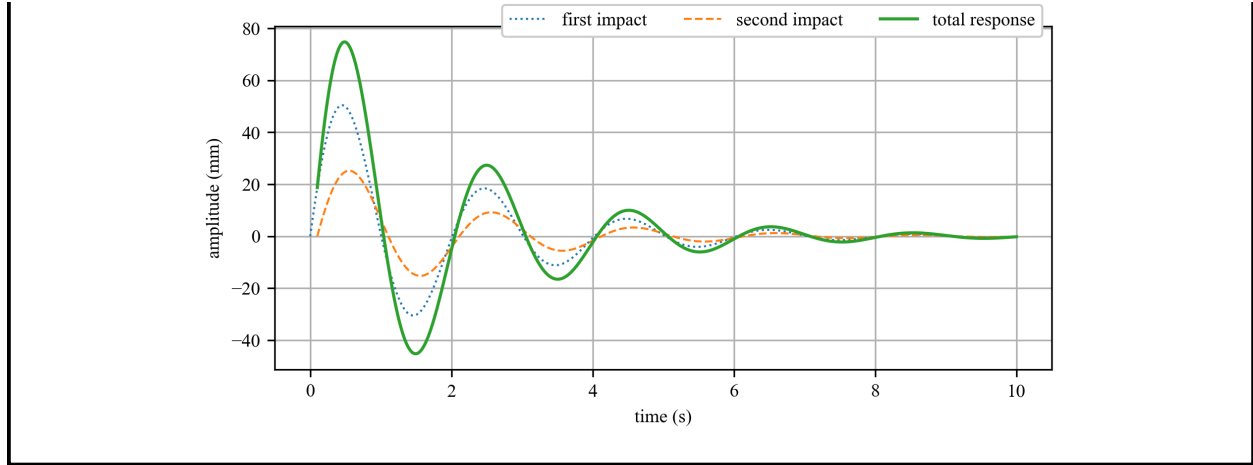
$$x_1(t) = \frac{0.2}{m\omega_d} e^{-\zeta\omega_n t} \sin(\omega_d t) \quad (45)$$

$$x_2(t) = \frac{0.1}{m\omega_d} e^{-\zeta\omega_n(t-\tau)} \sin(\omega_d(t-\tau)) \quad (46)$$

Next, using the knowledge that the systems are linear and that the Laplace transform of a linear combination of two transforms is the same as the linear transformation of these functions we can build the piecewise function:

$$x(t) = x_1(t) + x_2(t) = \begin{cases} \frac{0.2}{m\omega_d} e^{-\zeta\omega_n t} \sin(\omega_d t) & \text{if } t < \tau \\ \frac{0.2}{m\omega_d} e^{-\zeta\omega_n t} \sin(\omega_d t) + \frac{0.1}{m\omega_d} e^{-\zeta\omega_n(t-\tau)} \sin(\omega_d(t-\tau)) & \text{if } \tau \leq t \end{cases}$$

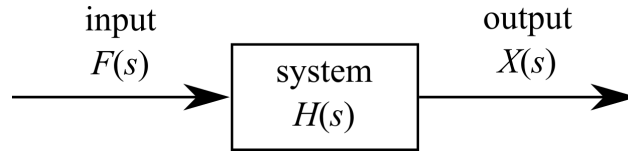
For the mass, damping, and stiffness values given above this can be plotted as:



4.4 Transform function for response to random inputs

4.4.1 Defining the transfer function $H(s)$

Again, consider the following system



For this system representation, $F(s)$ is the Laplace of the transform of the driving force and $H(s)$ is the Laplace transform of the response of the system $h(t)$.

We need to define transfer function $H(s)$ for a generic system. To do this let us show the reasoning behind the transfer function. **Here we will show that the output of any system ($x(t)$) can be related to the input of the system ($f(t)$) through a series of polynomial coefficients (a and b).** Consider the general n^{th} -order linear, time-invariant differential equation that governs the behavior of the dynamic system.

$$a_n \frac{d^n x(t)}{dt^n} + a_{n-1} \frac{d^{n-1} x(t)}{dt^{n-1}} + \dots + a_0 x(t) = b_m \frac{d^m f(t)}{dt^m} + b_{m-1} \frac{d^{m-1} f(t)}{dt^{m-1}} + \dots + b_0 f(t) \quad (47)$$

where $x(t)$ is the output and $f(t)$ is the input. Note that this is similar to the formulation we have had before for the EOM. Taking the Laplace transformation of both side of the above equation yields

$$a_n s^n X(s) + a_{n-1} s^{n-1} X(s) + \dots + a_0 X(s) + \text{initial condition involving } x(t) = (48) \\ b_m s^m F(s) + b_{m-1} s^{m-1} F(s) + \dots + b_0 F(s) + \text{initial condition involving } f(t)$$

It can be seen that this equation is a purely algebraic expression. If we assume the initial conditions to be zero, the equation reduces to the following:

$$(a_n s^n + a_{n-1} s^{n-1} + \dots + a_0) X(s) = (b_m s^m + b_{m-1} s^{m-1} + \dots + b_0) F(s) \quad (49)$$

we can rearrange the equation as

$$H(s) = \frac{X(s)}{F(s)} = \frac{b_m s^m + b_{m-1} s^{m-1} + \dots + b_0}{a_n s^n + a_{n-1} s^{n-1} + \dots + a_0} \quad (50)$$

where $H(s)$ is the **transfer function** that is defined as:

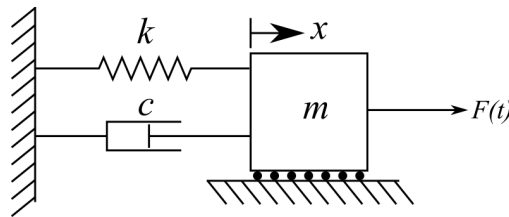
The ratio of the Laplace transforms of the output or response function to the laplace transform of the input or forcing function assuming zero initial conditions.

This can be rearranged to show that the output of the system $X(s)$ can be obtained if we know the input $F(s)$ and the transfer function $H(s)$:

$$X(s) = H(s) F(s) \quad (51)$$

4.4.2 Transfer Function method (Steady-State solution)

Considering the forced system:



that can be expressed as the equation of motion

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos(\omega t) \quad (52)$$

Here $F_0 \cos(\omega t)$, is used at the input but any input will develop the same transfer function as the transfer function is bounded to the system and not the input. From the #6 in the table for Laplace Transforms, we know that

$$\mathcal{L}[\cos(\omega t)] = \frac{s}{s^2 + \omega^2} \quad (53)$$

Therefore,

$$F(s) = \frac{F_0 s}{s^2 + \omega^2} \quad (54)$$

Ignoring the initial conditions, and therefore considering only the partial solution, and taking the Laplace transform of the EOM equation yields:

$$(ms^2 + cs + k)X(s) = \frac{F_0 s}{s^2 + \omega^2} \quad (55)$$

where s is the complex transform variable and $X(s)$ denotes the Laplace transform of the unknown function $x(t)$. Solving algebraically for the $X(s)$ yields:

$$X(s) = \frac{F_0 s}{(ms^2 + cs + k)(s^2 + \omega^2)} \quad (56)$$

Now that we have $F(s)$ and $X(s)$ we can obtain $H(s)$ as

$$H(s) = \frac{X(s)}{F(s)} = \frac{F_0 s}{(ms^2 + cs + k)(s^2 + \omega^2)} \cdot \frac{s^2 + \omega^2}{F_0 s} = \frac{1}{ms^2 + cs + k} \quad (57)$$

or

$$H(s) = \frac{1}{ms^2 + cs + k} \quad (58)$$

This ratio is termed the **transfer function of a system** and is an important tool in vibration analysis.

Recall that s is a complex number, if the value of s is restricted to lie along the imaginary axis in the complex plane (i.e., if $s = j\omega$), the transfer function becomes:

$$H(j\omega) = \frac{1}{m(j\omega)^2 + cj\omega + k} = \frac{1}{-m\omega^2 + cj\omega + k} \quad (59)$$

rearranging into a standard form yields:

$$H(j\omega) = \frac{1}{k - m\omega^2 + c\omega j} \quad (60)$$

recall that $j^2 = -1$. This is the **frequency response function of the system**. Therefore, it can be seen that the frequency response function of the system is the transfer function of the system evaluated along the imaginary axis $s = j\omega$. However, this expression contains imaginary values (that help to account for the phase in the system) and therefore can be challenging to work with. If we ignore

the phase component and only consider the amplitude $|H(j\omega)|$ of the response (the real portion of the equation) we get:

$$H(\omega) = |H(j\omega)| = \frac{1}{\sqrt{(k - m\omega^2)^2 + c^2\omega^2}} \quad (61)$$

where $|H(j\omega)|$ represents only the amplitude of the frequency response function. A more detailed derivation can be found in [Mechanical Vibrations - Rao, 6th edition]

To recap, for a single DOF damped spring-mass system the **transfer function** is:

$$H(s) = \frac{1}{ms^2 + cs + k} \quad (62)$$

And the **frequency response function** is:

$$H(j\omega) = \frac{1}{k - m\omega^2 + c\omega j} \quad (63)$$

While the amplitude of the frequency response is:

$$H(\omega) = |H(j\omega)| = \frac{1}{\sqrt{(k - m\omega^2)^2 + c^2\omega^2}} \quad (64)$$

4.4.3 Response to Random Inputs

The transfer and frequency response functions can be very useful for demeriting the response to random inputs. Up to this point we have solved for deterministic input.

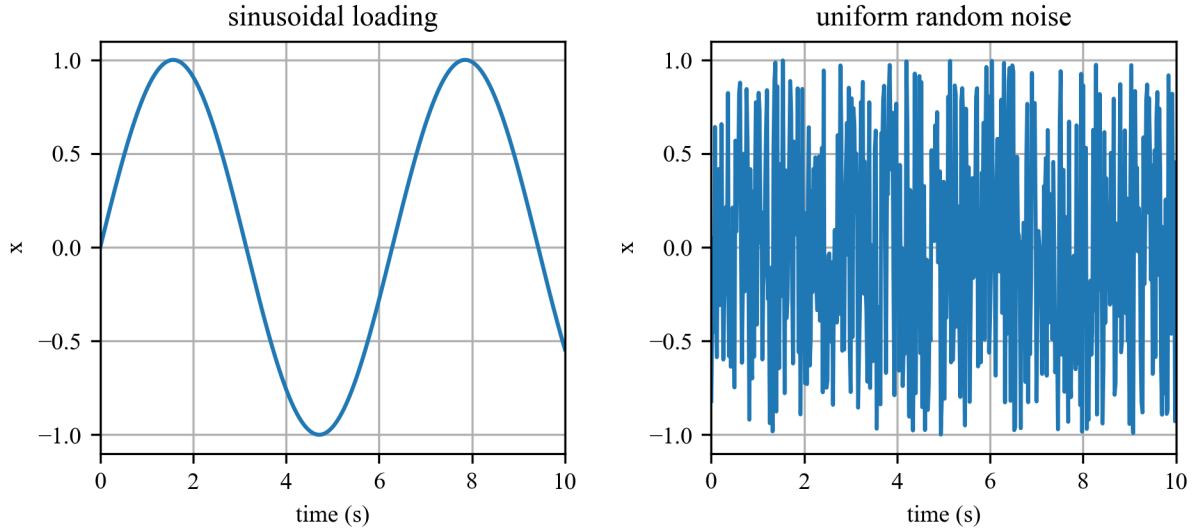
- **Deterministic**-For a known time t , the value of the input force $F(t)$ is precisely known.
- **Random** For a known time t , the value of the input force $F(t)$ is known only statistically.

To expand, a random signal is a signal with no obvious pattern. For these types of it is not possible to focus on the details of the input signal, as is done with a deterministic signal, rather the signal is classified and manipulated in terms of its statistical properties.

Randomness in vibration analysis can be thought of as the result of a series of results obtained from testing a system repeatability for various inputs under

varying conditions. In these cases, one record or time history is not enough to describe the system. Rather, an ensemble of various tests are used to describe how the system will respond to the various inputs.

First, let us consider two inputs, a deterministic input (typical sin wave), and a random input (white noise). These inputs are shown below.



One of the first factors to consider is the **mean of the random signal** $x(t)$, defined as:

$$E[x] = \bar{x} = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T x(t) dt \quad (65)$$

where T is the length in time of the data collected. However, for random signals we often want to consider signals with an average mean of zero (i.e. $\bar{x}(t) = 0$). Therefore, for signals not centered around zero we can obtain a zero centered signal if the signal is stationary and we subtract the mean value from \bar{x} from the signal $x(t)$. This can be written as:

$$x'(t) = x(t) - \bar{x} \quad (66)$$

where the $x'(t)$ is now centered around zero. As mentioned before, it is important to consider whether or not the input signals are stationary. A signal is **stationary** if its statistical properties (usually expressed by its mean) do not change with time. Here, it can be seen that for our inputs considered the signals are stationary if a long enough time period is considered.

Another important variable is **variance** (or mean-square value) of the random variable $x(t)$ defined as:

$$E[x^2] = \bar{x^2} = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T x^2(t) dt \quad (67)$$

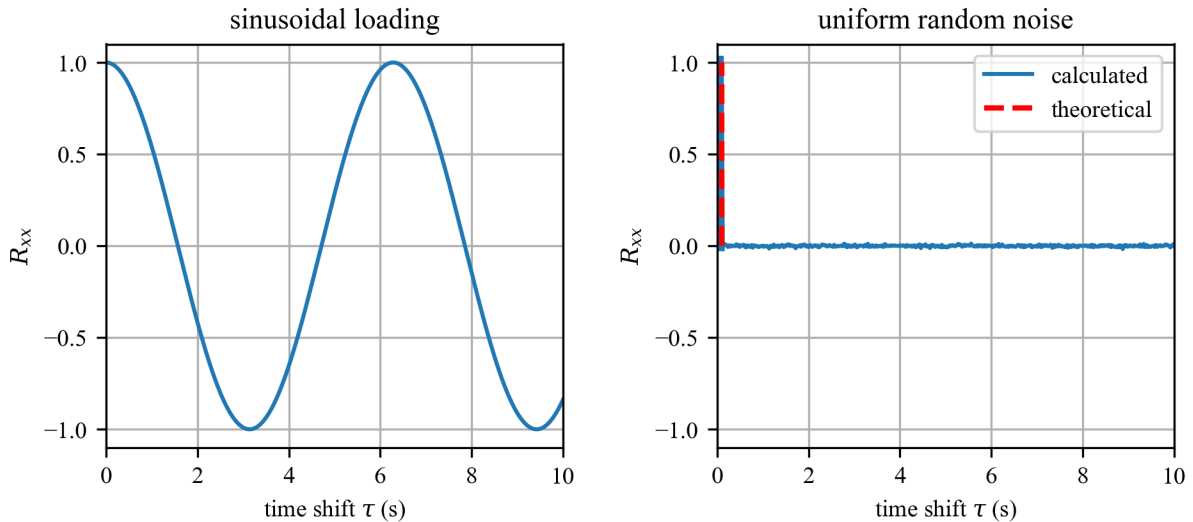
and provides a measure of the magnitude of the fluctuations in the signal $x(t)$. This gives an easy way to calculate the root-mean-square (RMS) of the signal as:

$$x_{\text{rms}} = \sqrt{\bar{x^2}} \quad (68)$$

Now, considering the actual signal, a important measure of interests is how fast the value of the variables change. This is important to understand as it address how long it takes to measure enough samples of the variable before a meaningful statistical value can be calculated. One way to quantify how fast the values of signal change is the **autocorrelation function**:

$$R_{xx}(\tau) = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T x(t)x(t + \tau) dt \quad (69)$$

The subscript xx denotes that this is a measure of the response for the variable xx , τ is the time difference between the values at which the signal $x(t)$ is sampled. The auto collation for the two inputs considered above are expressed below:



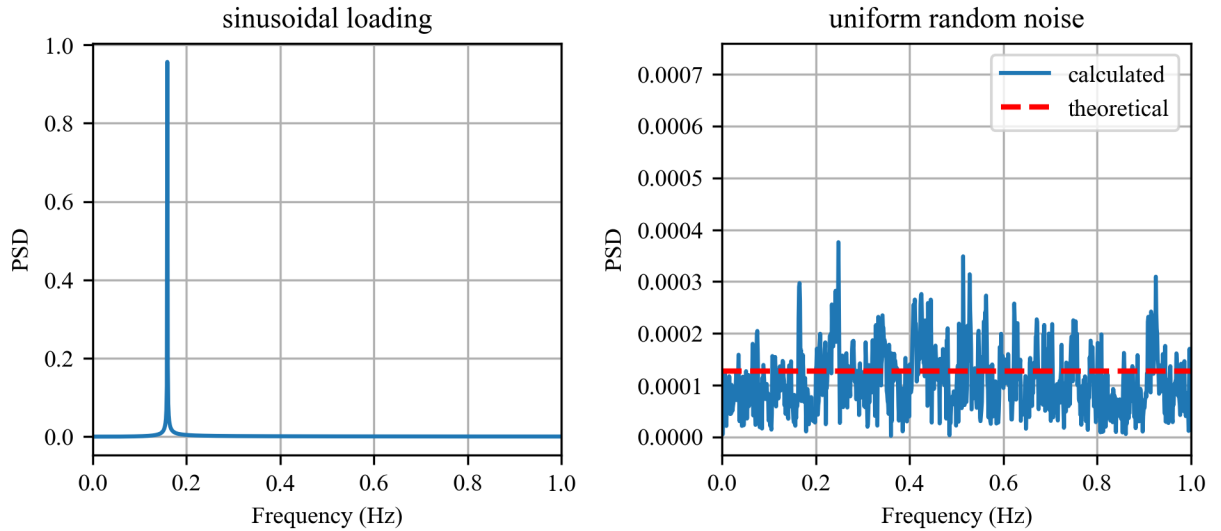
Note that the value of τ selected in the auto correlation function greatly affects its response for the sinusoidal input. This is because the values for the sinusoidal

are high correlated. To expand, the value at any time t is greatly effected by the values immediately before and after it. This is not the case for the random input where the signal is not correlated and therefore there is little difference in changing the value of τ on the response of the autocorrelation function.

Next, if we take the Fourier transform of the autocorrelation function we obtain the power spectral density (PSD) defined as:

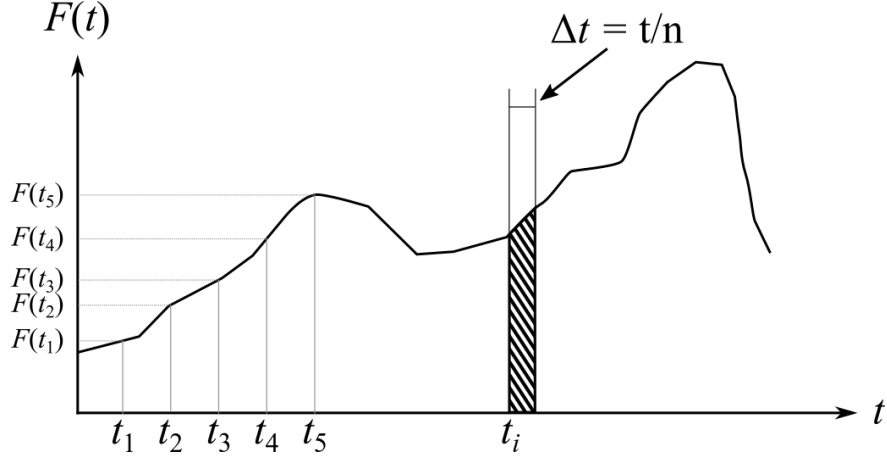
$$S_{xx}(\omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} R_{xx}(\tau) e^{-j\omega\tau} d\tau \quad (70)$$

where the integral of $R_{xx}(\tau)$ changes the real number τ into the frequency-domain value ω . The frequency spectrum (hence the S in $S_{xx}(\omega)$) for the two input cases considered are plotted below.



where the flat frequency response for the random input denotes that the random input is white noise input. A true white noise input would be perfectly flat, and is really just a theoretical concept.

Recall that S_{xx} is the spectrum of the response of the system. For the one-DOF system considered here, we can express an arbitrary input as a series of impulse inputs.



This knowledge, along with the frequency response function can be used to relate the spectrum of the input $S_{ff}(\omega)$ to the output through the transfer function as:

$$S_{xx}(\omega) = |H(j\omega)|^2 \left[\frac{1}{2\pi} \int_{-\infty}^{\infty} R_{ff}(\tau) e^{-j\omega\tau} d\tau \right] \quad (71)$$

This can also be expressed in symbolic form as

$$S_{xx}(\omega) = |H(j\omega)|^2 S_{ff}(\omega) \quad (72)$$

where R_{ff} denotes the autocorrelation function of $F(t)$ and S_{ff} denotes the PSD of the forcing function $F(t)$. The notation $|H(j\omega)|^2$ is the square of the magnitude of the complex frequency response function. A more detailed derivation can be found in [Engineering Vibrations, Inman (2001)], [Random Vibrations, Spectral & Wavelet Analysis, Newland (1993)], but here it is more important to study the results rather than the derivations.

Another useful quantity to consider is the expected value of the output for a given input. The expected value $E[x]$, or \bar{x} , is expressed as:

$$E[x] = \lim_{T \rightarrow \infty} \int_0^T \frac{x(t)}{T} dt \quad (73)$$

Additionally, the mean-square value can be directly related to the PSD function as

$$E[x^2] = \bar{x}^2 = \int_{-\infty}^{\infty} |H(j\omega)|^2 S_{ff}(\omega) d\omega \quad (74)$$

Moreover, for a constant input $S_{ff}(\omega) = S_0$ in terms of the frequency domain, the mean-square value can be expressed as:

$$E[x^2] = \bar{x}^2 = S_0 \int_{-\infty}^{\infty} |H(j\omega)|^2 d\omega \quad (75)$$

After inspecting the above equation, it becomes clear that to obtain the square of the expected value, a solution for $\int_{-\infty}^{\infty} |H(j\omega)|^2$ must be obtained. For cases where $S_{ff}(\omega) = S_0$ and as such $S_{ff}(\omega)$ can be pulled out of the integral, these integrals have been solved [Random Vibrations, Spectral & Wavelet Analysis, Newland (1993)]. For example, given $\int_{-\infty}^{\infty} |H(j\omega)|^2 d\omega$:

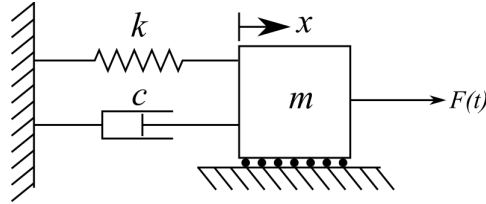
$$\int_{-\infty}^{\infty} \left| \frac{B_0}{A_0 + j\omega A_1} \right|^2 d\omega = \frac{\pi B_0^2}{A_0 A_1} \quad (76)$$

and

$$\int_{-\infty}^{\infty} \left| \frac{B_0 + j\omega B_1}{A_0 + j\omega A_1 - \omega^2 A_2} \right|^2 d\omega = \frac{\pi(A_0 B_1^2 + A_2 B_0^2)}{A_0 A_1 A_2} \quad (77)$$

When combined with the equation $E[x^2] = S_0 \int_{-\infty}^{\infty} |H(j\omega)|^2 d\omega$, these integrals allow for the easy calculation of the expected values.

Example 4.3 Considering the forced system:



Set the forcing function to be $F_0 \sin(\omega t)$ and calculate the transfer function.

Solution: That can be expressed as the equation of motion

$$m\ddot{x} + c\dot{x} + kx = F_0 \sin(\omega t) \quad (78)$$

From the #6 in the table for Laplace Transforms, we know that

$$\mathcal{L}[\sin(\omega t)] = \frac{\omega}{s^2 + \omega^2} \quad (79)$$

Therefore,

$$F(s) = \frac{F_0 \omega}{s^2 + \omega^2} \quad (80)$$

Ignoring the initial conditions and taking the Laplace transform of the EOM equation yields:

$$(ms^2 + cs + k)X(s) = \frac{F_0\omega}{s^2 + \omega^2} \quad (81)$$

Solving algebraically for the $X(s)$ yields:

$$X(s) = \frac{F_0\omega}{(ms^2 + cs + k)(s^2 + \omega^2)} \quad (82)$$

Now that we have $F(s)$ and $X(s)$ we can obtain $H(s)$ as

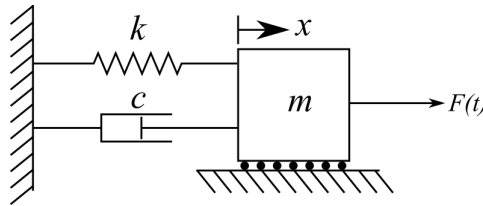
$$H(s) = \frac{X(s)}{F(s)} = \frac{F_0\omega}{(ms^2 + cs + k)(s^2 + \omega^2)} \cdot \frac{s^2 + \omega^2}{F_0\omega} = \frac{1}{ms^2 + cs + k} \quad (83)$$

or

$$H(s) = \frac{1}{ms^2 + cs + k} \quad (84)$$

This is identical to the solution obtained using $F_0 \cos(\omega t)$ as would be expected because the transfer function is related to the system and not to the input.

Example 4.4 Consider the following system



Calculate the PSD of the response $x(t)$ given that the PSD of the applied force $S_{ff}(\omega)$ is white noise.

Solution: From the system we know that the EOM is

$$m\ddot{x} + c\dot{x} + kx = F(t) \quad (85)$$

The frequency response function for this system is

$$H(j\omega) = \frac{1}{k - m\omega^2 + c\omega j} \quad (86)$$

while the amplitude of the response is:

$$H(\omega) = |H(j\omega)| = \frac{1}{\sqrt{(k - m\omega^2)^2 + c^2\omega^2}} \quad (87)$$

Applying the equation that relates $S_{ff}(\omega)$ to $S_{xx}(\omega)$ we get:

$$S_{xx}(\omega) = |H(j\omega)|^2 S_{ff}(\omega) = \left| \frac{1}{k - m\omega^2 + c\omega j} \right|^2 S_{ff}(\omega) \quad (88)$$

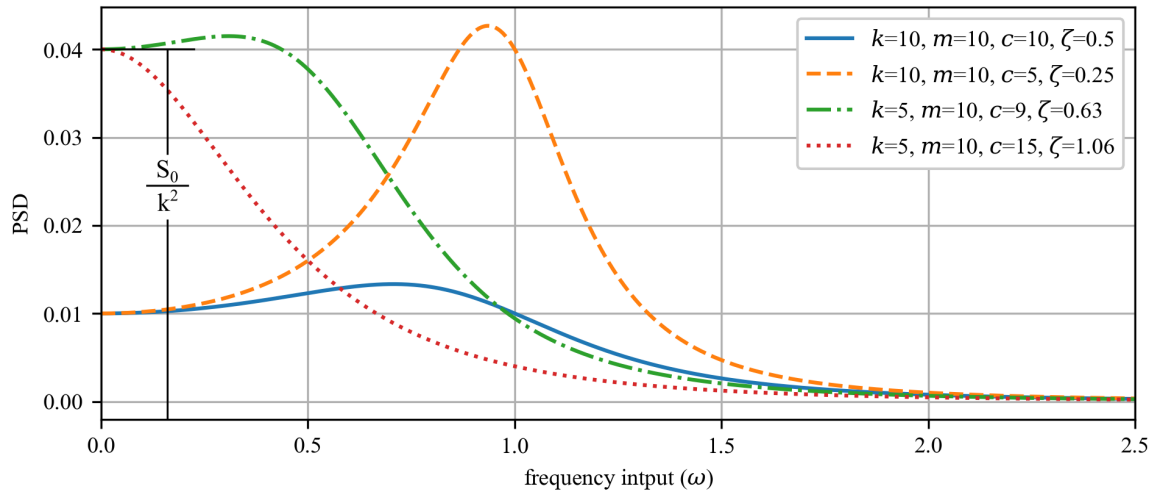
White noise means the forcing function $S_{ff}(\omega)$ is constant across the frequency spectrum, therefore, $S_{ff}(\omega) = S_0$. Additionally as:

$$|H(j\omega)|^2 = \left| \frac{1}{k - m\omega^2 + c\omega j} \right|^2 = \frac{1}{(k - m\omega^2)^2 + c^2\omega^2} \quad (89)$$

where the absolute value is the amplitude of the system. Therefore, we obtain:

$$S_{xx}(\omega) = |H(j\omega)|^2 S_0 = \frac{1}{(k - m\omega^2)^2 + c^2\omega^2} S_0 = \frac{S_0}{(k - m\omega^2)^2 + c^2\omega^2} \quad (90)$$

Using various values for the elements in the system, the PSD for the system considered looks like:



Example 4.5 For the same system as example 1, calculate the mean-square response of the system.

Solution: Again, as the forcing function $S_{ff}(\omega)$ is constant across the frequency spectrum $S_{ff}(\omega) = S_0$ the mean-square response can be calculated as:

$$E[x^2] = \bar{x}^2 = S_0 \int_{-\infty}^{\infty} |H(j\omega)|^2 d\omega \quad (91)$$

Using the already tabulated response:

$$\int_{-\infty}^{\infty} \left| \frac{B_0 + j\omega B_1}{A_0 + j\omega A_1 - \omega^2 A_2} \right|^2 d\omega = \frac{\pi(A_0 B_1^2 + A_2 B_0^2)}{A_0 A_1 A_2} \quad (92)$$

and the frequency response function:

$$H(j\omega) = \frac{1}{k - m\omega^2 + c\omega j} \quad (93)$$

when $B_0 = 1$, $B_1 = 0$, $A_0 = k$, $A_1 = c$, and $A_2 = m$. Therefore, using the tabulated expression we can show that:

$$E[x^2] = S_0 \frac{\pi m}{k c m} = \frac{S_0 \pi}{k c} \quad (94)$$

Table of Laplace Transforms for Vibrations

This is a partial lists of important Laplace transforms for vibrations that assumes zero initial conditions, $0 < t$, and $\zeta < 1$.

$f(t)$	$\mathcal{L}[f(t)] = F(s)$		$f(t)$	$\mathcal{L}[f(t)] = F(s)$	
$\delta(t)$	1	(1)	$\frac{1}{\omega^3}(\omega t - \sin(\omega t))$	$\frac{1}{s^2(s^2 + \omega^2)}$	(17)
$\delta(t - t_0)$	e^{-st_0}	(2)	$\frac{1}{2\omega^3}(\sin(\omega t) - \omega t \cos(\omega t)) \dots$	$\frac{1}{(s^2 + \omega^2)^2}$	(18)
1	$\frac{1}{s}$	(3)	$\frac{t}{2\omega} \sin(\omega t)$	$\frac{s}{(s^2 + \omega^2)^2}$	(19)
e^{at}	$\frac{1}{s - a}$	(4)	$t \sin(\omega t)$	$\frac{2\omega s}{(s^2 + \omega^2)^2}$	(20)
$\sin(\omega t)$	$\frac{\omega}{s^2 + \omega^2}$	(5)	$t \cos(\omega t)$	$\frac{s^2 - \omega^2}{(s^2 + \omega^2)^2}$	(21)
$\cos(\omega t)$	$\frac{s}{s^2 + \omega^2}$	(6)	$e^{at} \sin(\omega t)$	$\frac{\omega}{(s - a)^2 + \omega^2}$	(22)
$\sinh(\omega t)$	$\frac{\omega}{s^2 - \omega^2}$	(7)	$e^{at} \cos(\omega t)$	$\frac{s - a}{(s - a)^2 + \omega^2}$	(23)
$\cosh(\omega t)$	$\frac{s}{s^2 - \omega^2}$	(8)	$e^{at} \sinh(\omega t)$	$\frac{\omega}{(s - a)^2 - \omega^2}$	(24)
$\frac{1}{\omega^2}(1 - \cos(\omega t))$	$\frac{1}{s(s^2 + \omega^2)}$	(9)	$e^{at} \cosh(\omega t)$	$\frac{s - a}{(s - a)^2 - \omega^2}$	(25)
$\frac{1}{\omega_d} e^{-\zeta \omega t} \sin(\omega_d t)$	$\frac{1}{s^2 + 2\zeta \omega s + \omega^2}$	(10)	$\frac{1}{\omega_2} \sin(\omega_2 t) - \frac{1}{\omega_1} \sin(\omega_1 t) \dots$	$\frac{\omega_1^2 - \omega_2^2}{(s^2 + \omega_1^2)(s^2 + \omega_2^2)}$	(26)
$1 - \frac{\omega}{\omega_d} e^{-\zeta \omega t} \sin(\omega_d t + \phi), \phi = \cos^{-1}(\zeta) \dots$	$\frac{\omega^2}{s(s^2 + 2\zeta \omega s + \omega^2)}$	(11)	$\cos(\omega_2 t) - \cos(\omega_1 t)$	$\frac{s(\omega_1^2 - \omega_2^2)}{(s^2 + \omega_1^2)(s^2 + \omega_2^2)}$	(27)
$\frac{t^{n-1}}{(n-1)!}, n = 1, 2, \dots$	$\frac{1}{s^n}$	(12)	$e^{at} f(t)$	$F(s - a)$	(28)
$t^n, n = 1, 2, \dots$	$\frac{n!}{s^{n+1}}$	(13)	$f(t - a)\Phi(t - a)$	$e^{-as}F(s)$	(29)
$t^n e^{\omega t}, n = 1, 2, \dots$	$\frac{n!}{(s - \omega)^{n+1}}$	(14)	$\Phi(t - a)$	$\frac{e^{-as}}{s}$	(30)
$\frac{1}{\omega}(1 - e^{-\omega t})$	$\frac{1}{s(s + \omega)}$	(15)	$f'(t)$	$sF(s) - f(0)$	(31)
$\frac{1}{\omega^2}(e^{-\omega t} + \omega t - 1)$	$\frac{1}{s^2(s + \omega)}$	(16)			