Cam Motion Case Studies #1 and # 2

Problem/Opprtunity:

At an operating speed of 150 to 160 rpm, Cam Motion #1 causes the cam follower to leave the cam surface unless excessive air pressure is applied to the air spring that maintains follower contact. An operating speed of 175 rpm or better is the target speed for this upgrade. Cam Motion #2 operates acceptably at 150 to 160 rpm but is being addressed because it works in conjunction with Cam Motion #1 and the best possible performance at the higher target speed is desired.

Results Summary:

The application of motions generated with our polynomial techniques combined with the strategic recognition of the functional requirements for each motion yielded optimized kinematic characteristics at speeds exceeding the target production rate. In fact, the client elevated their target speed to 200 rpm after implementing these and other redesigned cam motions. Additionally, we presented a Polynomial Motion Design Seminar for their engineering staff. They have subsequently abandoned their old motion design practices and now use polynomial motions exclusively.

Analysis and Redesign Strategy:

Both cams are open face plate cams with oscillating follower arms producing an angular output. Cam #1 produces a linear stroke through linkage as its end result while Cam #2 produces a magnified angular output. Motion #1 engages a slotted shaft onto the stub of an indexing nest. Motion #2 rotates the stub 90E and then returns it to the original orientation. The nests index through the station during the period from 0E to 150E on the kinematic graph. The nest index period was not to be changed.

It is desired to engage the slotted shaft onto the nest stub quickly so that the motion periods for the 90E turning can be of reasonable length. The linear stroke from Cam #1 is presently 0.187 inches. The existing design approach minimized this stroke and engages the stub with a short 20E Modified Trapezoid motion. This startegy accomplishes the quick engagement but produces high accelerations resulting in the follower jump situation and the demand for high air spring pressure. The “Kinematic Comparison – Cam Motion Case Study #1” displays the present Modified Trapezoid motion’s velocity and acceleration characteristics at 175 rpm in blue.

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Cam Motion Case Studies #1 and #2 (continued)

In order to lower the peak accelerations from Motion #1, we re-examined the functional requirements for this motion. First, the slotted shaft must be clear of the nest stub during the nest index. Second, the slotted shaft must be sufficiently engaged with the nest stub when the stub rotation motion begins and ends. Our approach was to increase the slotted shaft linear displacement to the limits of the existing slide and linkage boundaries and develop a single polynomial motion that eliminated the dwell during disengagement. This motion was also overlapped with the beginning and end of the stub rotation motions by 10E. Instead of having two distinct engagement and disengagement motions of 20E each, we now have a single polynomial motion whose duration is 210E. The new velocity and acceleration characteristics at 175 rpm are shown in red on the graphs for Case Study #1. The peak acceleration is now 8.53% of the Modified Trapezoid's magnitude that caused the follower jump problem. Using a lower air spring pressure, the permissible operating speed of the redesigned Motion #1 was calculated to be well in excess of 225 rpm. The peak velocity of the new motion is 39.15% of the old motion's value. The performance of the new Cam #1 was a true paradigm shift from the old version and this function was no longer a speed limitation for the target production rate.

Because Cam #2 was operating fairly well, its total motion period for rotation and return was left at 170E for the initial redesign examination. It turned out that this worked quite well. The function of the stub rotation was to snap off a portion of the product that was used exclusively for holding purposes during previous operations. The breakage always occurred by one half of the 90E rotation. The balance of the rotation simply completed the 90E oscillation and returned the mechanism to its original starting position. Therefore, the 10E dwell between the forward rotation and the return was not functionally required. A single polynomial motion was developed to produce both the forward and return motions. The acceleration and velocity characteristics of the old and new motions are displayed in the Case Study#2 graphs.

The resulting performance for the Cam Motion #2 was almost as dramatic as Cam #1, providing a smooth, vibration free oscillation of the stub and all associated linkage. Its maximum operating speed was also in excess of 225 rpm.

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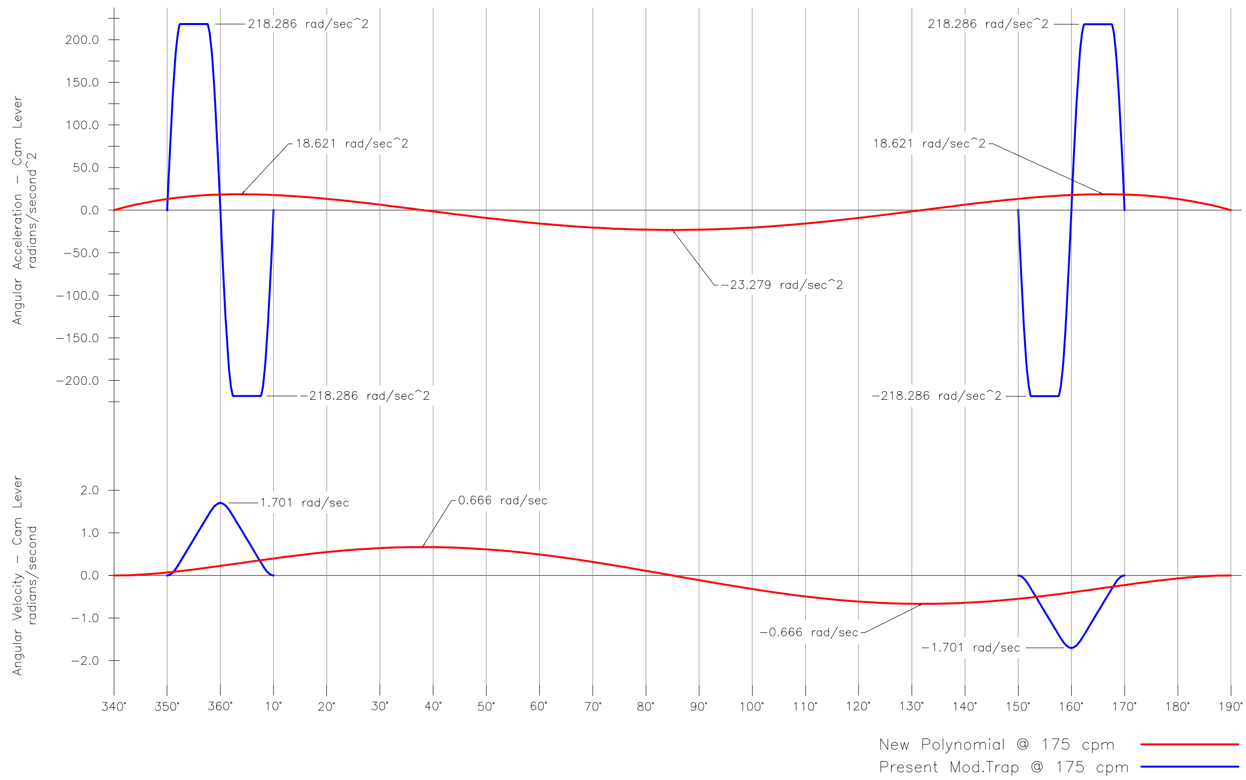
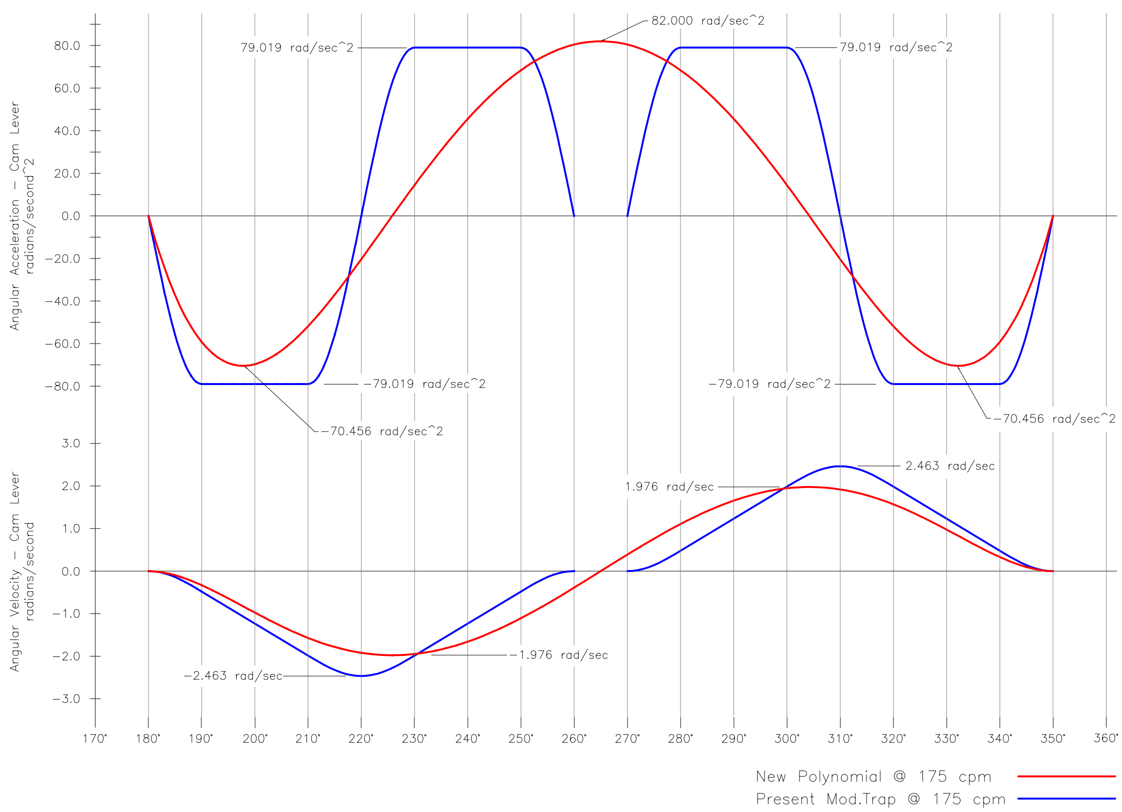


Kinematic Comparison – Cam Motion Case Study #1



Kinematic Comparison – Cam Motion Case Study #2

Cam Motion Case Study #3

Problem/Opportunity:

As part of a program to increase operating speed from 150 to 175 rpm, the time allowed to effect a product transfer had to be increased to make certain that the transfer was properly accomplished. This meant that the dwell segment of the cam operating the shuttle bar must be increased at the transfer end of the shuttle stroke. Other mechanism constraints dictated that the shuttle motion periods (forward and return) be shortened by 10E each, thus adding 20E to the critical dwell period. The current motions were each 145E in duration and utilized the Modified Trapezoid acceleration form. The performance of the shuttle at 150 to 160 rpm was judged to be acceptable. Therefore, the motion duration reduction was considered to be an appropriate action.

Results Summary:

A polynomial motion with as asymmetric acceleration function was developed as a replacement for the Modified Trapezoid motion. The new cam performed better at 175 rpm than the present cam did at a speed of 150 rpm. This fact was confirmed by a high speed video study. Significantly less vibration occurred when entering the transfer dwell period. Vibration from the motion was one of the factors causing transfer problems in the past.

Analysis and Redesign Strategy:

The attached “Kinematic Comparison – Cam Motion Case Study #3” displays the acceleration and velocity characteristics for the forward motion of both the old Modified Trapezoid at 150 rpm and the new polynomial at 175 rpm. The cam is an open face plate cam driving an oscillating follower arm that produces the linear shuttle stroke. An air spring is used to maintain follower contact with the cam surface. It was decided to favor the start of the motion as this was the area where the air spring force was required to prevent follower jump. Note that the acceleration and deceleration values for the polynomial are larger than the Modified Trapezoid acceleration form. The peak velocity is also higher. The fact that the new polynomial at the higher speed outperforms the existing cam at the present rate appears, on the surface, to defy conventional wisdom. In reality, the only feature that the Modified Trapezoid can claim is its relatively low peak acceleration. Overall performance is a balance between acceleration, velocity, input torque and vibrational characteristics. Polynomial motion techniques provide a powerful and creative tool to address the most challenging cam and servo motion requirements.

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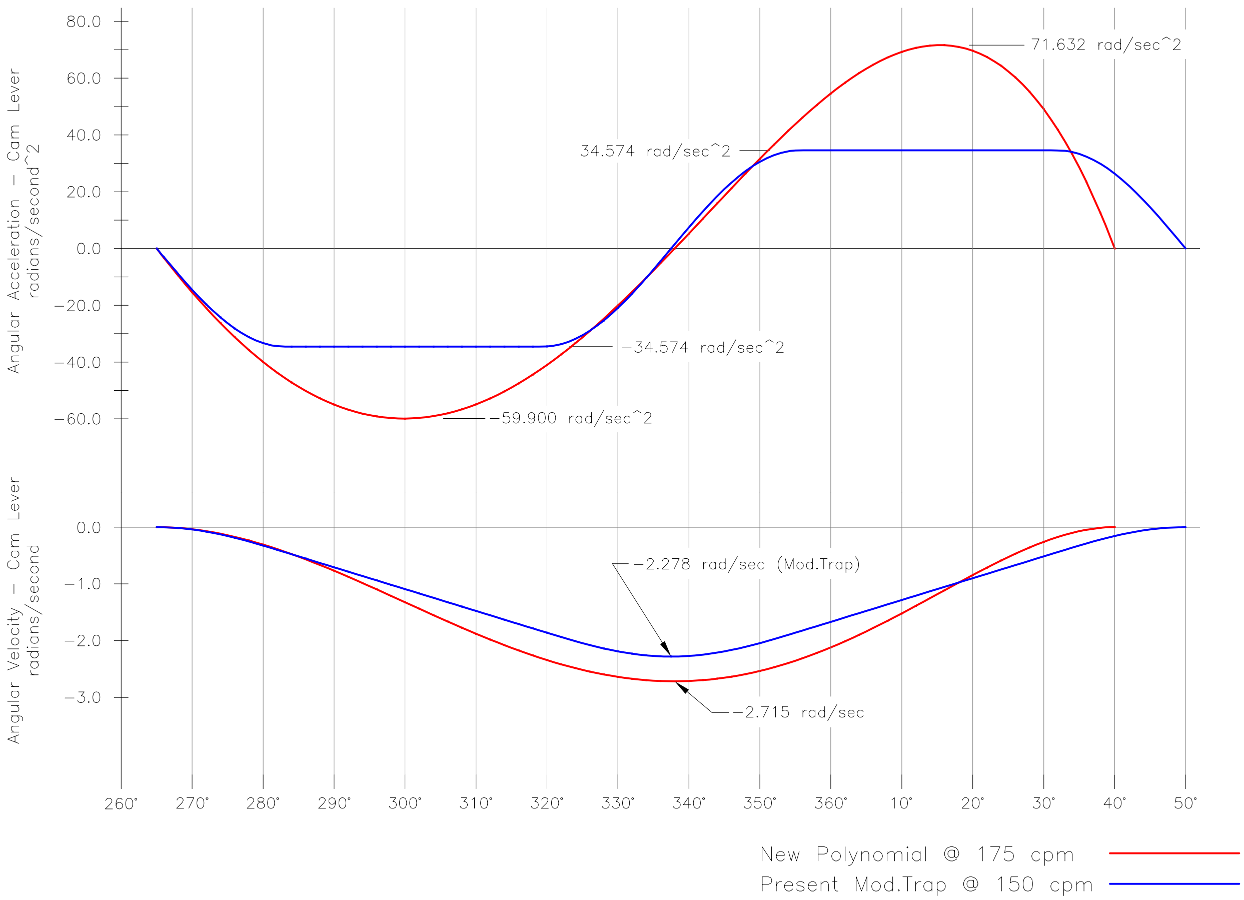

Kinematic Comparison – Cam Motion Case Study #3

Cam Motion Case Study #4

Problem/Opportunity:

The function of one cam mechanism on an indexing linear conveyor assembly machine is to locate each flight platen perpendicular to the conveyor center line when it is on the end (turn-around) of the machine. The action of this cam is abrupt and makes noise as it impacts the platen when running at the current speed of 150 rpm. The target speed for the upgrade project is 175 rpm.

Result Summary:

Analysis of the clearances required between the turning flight platens and the locating bar led to the development of a single rise/return polynomial cam motion to replace the three individual Modified Trapezoid motions used by the current cam. When compared at the same speed, the new motion reduces the peak acceleration magnitude by 90.86%. The peak velocity is reduced by 62.68%. The new cam's performance was incredible and led the client to elevate their target speed to 200 rpm.

Analysis and Redesign Strategy:

The attached "Kinematic Comparison – Cam Motion Study #4" shows the new polynomial motion and the existing Modified Trapezoid motions at the target speed of 175 rpm. A functional review of the station produced two basic requirements for the redesigned motion. First, the locator bar should be in dwell against the flight platen from 190E to 320E. The operation being performed requires that the platen is positively oriented during this period. Second, the locator bar must not interfere with the platens as they rotate around the end radius of the conveyor.

A detailed study of the platen clearance envelope showed that an ideal rise/return polynomial motion easily provides the necessary clearance using the locator bar dwell as presently defined with the existing locator bar stroke. Additionally, the old design uses two motions to advance the bar to the platen. The second short motion is intended to provide a "soft touch" of the bar to the platen. Examining the velocity comparison of this motion that occurs between 180E and 190E, this idea was really misguided as the velocity profile of the polynomial is far less than the Modified Trapezoid during this time. The resulting performance of the new polynomial motion confirms the importance of a practical and basic understanding of the mechanism requirements and that polynomial techniques provide a flexible and powerful tool for creating solid solutions for both cam and servo motion needs.

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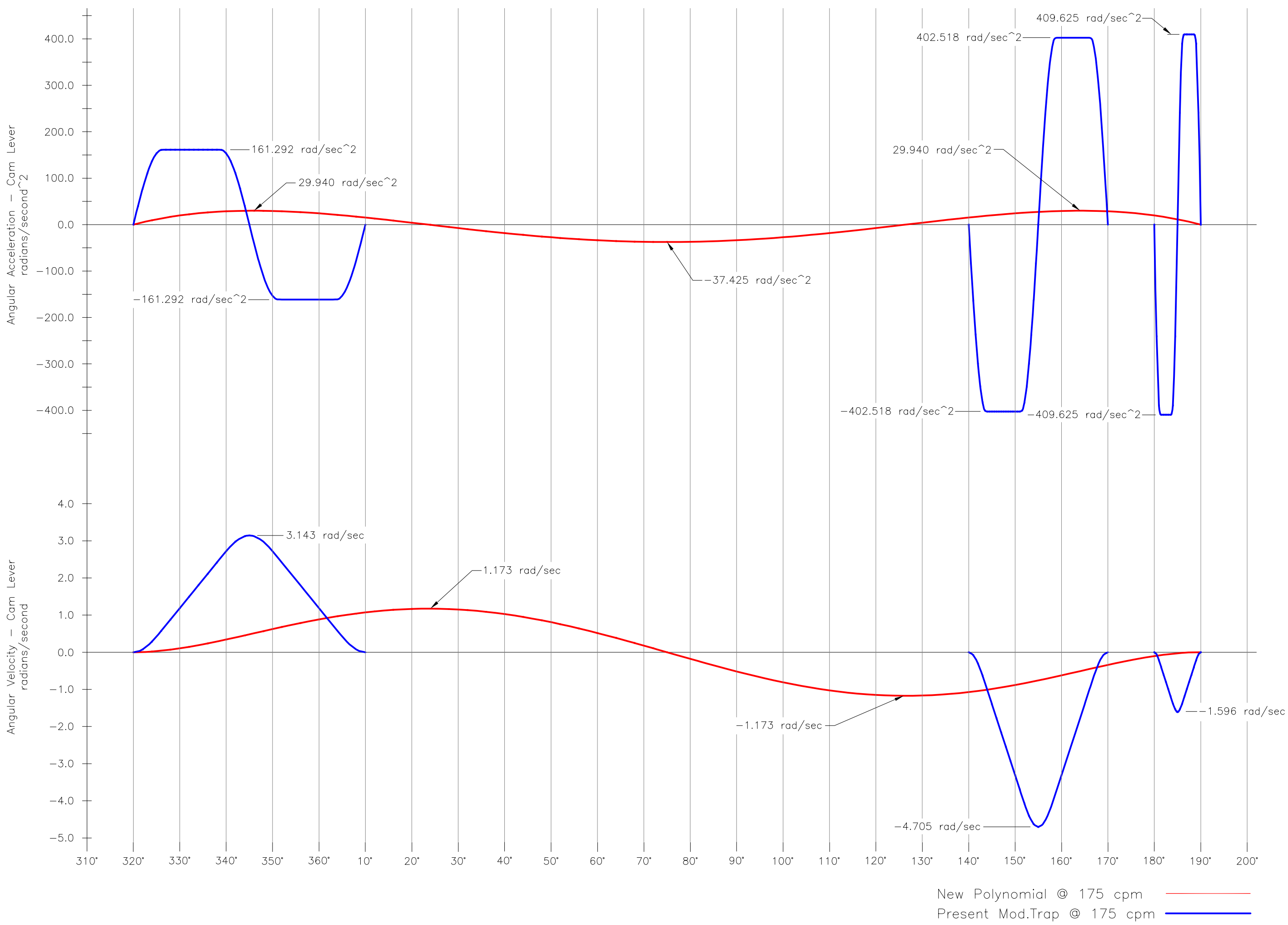

Kinematic Comparison – Cam Motion Case Study #4

Servo Motion Case Study #5

Problem/Opportunity:

A plurality of cylindrical products must be transferred from a staging area and deposited into containers. The product is somewhat compliant and easily damaged dictating minimum gripper finger pressure. The total pick and place cycle time is 4.5 seconds with 2.0 seconds available for the actual product placement. Optimum motion characteristics are desired to maintain proper control of the delicate product during the transfer. The weight of the pick-head is 215 pounds with a total carriage weight of 520 pounds. A servo motor drives a sprocket and timing belt as a rotary to linear converter for each axis. The horizontal displacement of the pick-head is 46.0 inches and a minimum vertical displacement of 2.75 inches is required to clear the staging nests.

Result Summary:

An eighth order polynomial for the vertical motion was applied in conjunction with a sixth order polynomial for the horizontal motion. The vertical displacement was increased to facilitate the use of the single polynomial for the vertical movement to minimize acceleration while providing the necessary nest clearance. Absolutely no vibration was felt in the mechanism frame at operating speed.

Analysis and Redesign Strategy:

The attached page shows the kinematics for each axis as well as the resultant path for one of the products. A 6.0 inch total stroke for the vertical motion and the beginning and ending offsets between the two motions were determined by experimentation so that the path nicely cleared the staging nests. The acceleration of the lift was favored by offsetting the peak displacement of the vertical axis towards the deposit point. The product positions indicated for the resultant path are at 0.1 second intervals.

A servo controller with a table driven or “cam” capability was employed to produce these motions. The results are far superior to any conventional velocity ramping motion that is often utilized for this type of mechanism, even if the motion overlaps between the two axes were still used.

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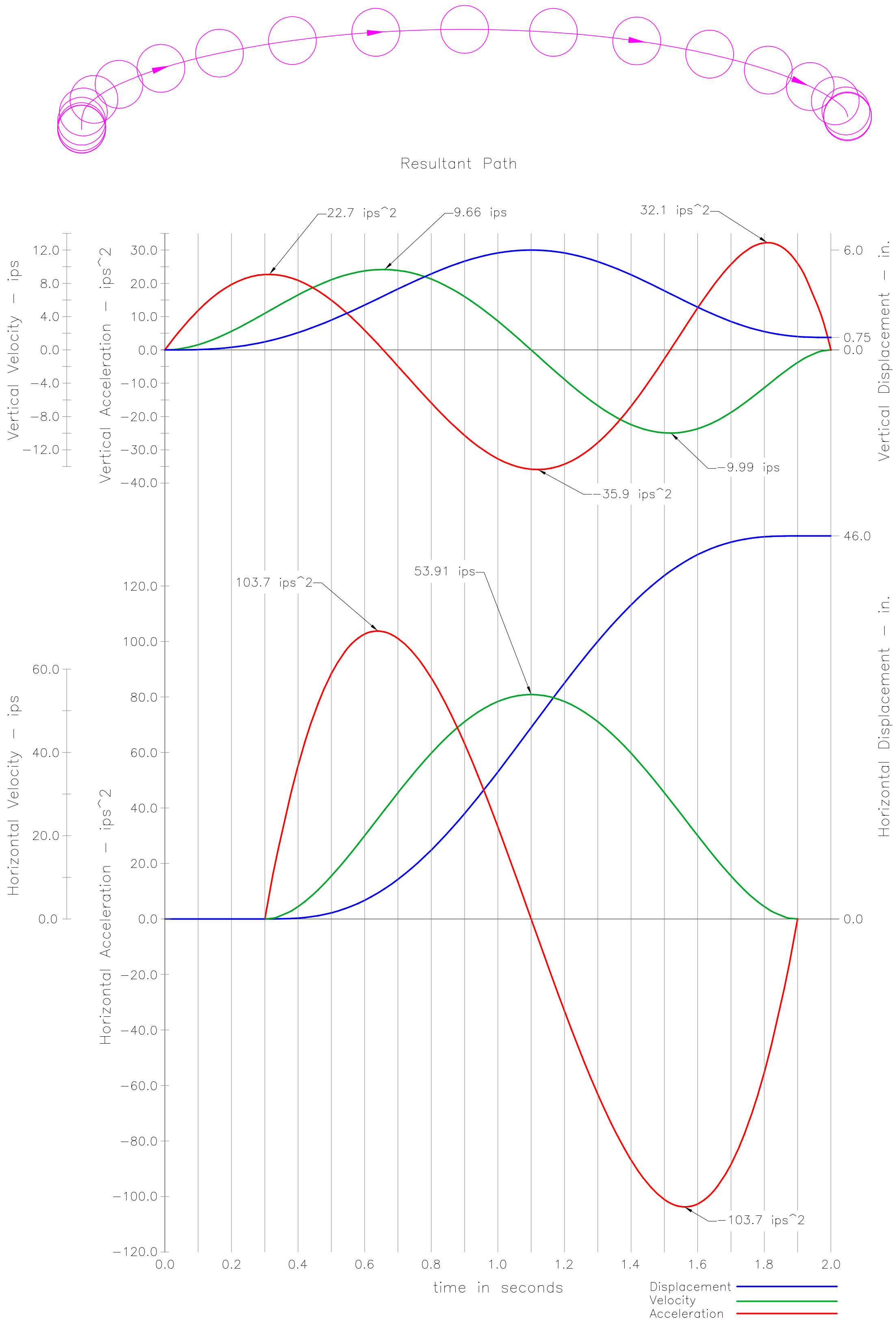

Case Study 5 – Pick and Place Kinematics