THERMODYNAMIC TREATMENT OF TUG-&-TWIST TECHNOLOGY:

PART 1 ... THERMODYNAMIC TUGGER DESIGN

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ABSTRACT
Especially applicable to arthrobots involving human interaction in light assembly and other delicate manipulation tasks, the new technical field variously called rubbertronics or pneumoelastics is here designated as tug-&-twist technology. Part 1 of this paper is concerned only with linear tension actuators or tuggers.

Such devices have generally been analyzed and designed by means of localized forces and stresses. Instead, the treatment here is based upon the application of thermodynamic principles and in particular the use of a complementary energy function in the form of a generalized ENTHALPY. In this way, common features of all past, present and future tugger designs are best revealed.

THE RISE OF ARTHROBOTS
Until the early 1970’s, multi-d.o.f industrial robots utilized a variety of coordinate motions: spherical (e.g. UNIMATE); cylindrical (e.g. VERSATRAN); cartesian, and others. But over the last 25 years, the jointed-arm configuration of Figure 1 became widespread (e.g. PUMA), wherein all degrees-of-freedom are joint rotations. This has meant that torque motors could power each and every axis at which uniform shaft angle encoders could then measure joint angles to provide closed-loop control. This author designates such specialized configurations as arthrobots and Part 1 of this paper is concerned with the use of tuggers to drive such systems, while Part 2 demonstrates how twistors can serve both as flexural bearings and as motors at each joint.

THREE TUGGERS
In 1984, Bridgestone Corporation announced a class of arthrobots employing pairs of antagonistic tuggers to move each joint in emulation of human musculature (Burgam, 1984). Their SOFTARM is illustrated in Figure 2 and their proprietary tugger employed was called the RUBBERTUATOR, with a typical data sheet as shown in Figure 4 and having the appearance as indicated in Figure 3(c). Its features are protected by several Japanese and US Patents (Takagi and Sakaguchi, 1986; Sakaguchi and Tershama, 1987).

However, much earlier in 1969, John Yarlett had formed Trish Energetics, Inc., to manufacture the first commercially-successful tugger as the AMPFLEX actuator, with the data sheet of Figure 5 and the schematic as in Figure 3(a). Its design was covered by a 1972 US Patent (Yarlett, 1972) and a 1975 Japan Patent (Yarlett, 1975).

Then as the successor enterprise to Trish Energetics, the author’s own company, Dynacyle Corporation, improved upon Yarlett’s design in 1972 to produce a high-pressure air-motor driven by DYNAFLEX tuggers, having the data sheet of Figure 6 and as illustrated in Figure 3(b). This particular design series ultimately led to US and Canadian Patents (Paynter, 1988; Paynter 1992).

These three types of tuggers have all been commercially produced and sold in many sizes and in large quantities. While many other patented tugger-designs now exist, only the three models of Figure 3 are discussed in this paper.

THERMODYNAMIC APPROACH TO DESIGN
John Yarlett and most other researchers designed their variant form of tuggers primarily on the basis of detailed analyses of force distributions. However, such procedures are unduly complex and fail to suggest alternative, improved geometries. A more fruitful approach is based upon the First Law of Thermodynamics, recognizing that tuggers represent a near-reversible conversion of fluid available energy into mechanical energy. This writer has employed such methods for tugger design during the last 25 years.

TUGGER STORED ENERGY
This approach begins with the First Law statement:

$$dE = P \cdot dV - F \cdot dX$$

(1)

where ENERGY \( E = E(V, X) \) represents a constraint between PRESSURE, P, VOLUME, V, FORCE, F, and STROKE, X. However, here we are interested in the situation where \( P \) and \( X \) are the proper input variables with \( V \) and \( F \) the corresponding output variables. For these conditions, we must perform a Legendre
transformation upon ENERGY E to form a complementary energy or generalized ENTHALPY H, as next outlined.

TUGGER ENTHALPY

Given \( E(V,X) \), the function \( H(P,X) \) is found from the familiar thermodynamic relation, taking due account of signs:

\[
\begin{align*}
    H & = E - PV = H(P,X) \quad (2a) \\
    dH & = -V \, dP - F \, dX \quad (2b)
\end{align*}
\]

From this relation we then obtain the outputs

\[
\begin{align*}
    \text{VOLUME} \ V & = -\frac{\partial H(P,X)}{\partial P} = V(P,X) \quad (3a) \\
    \text{FORCE} \ F & = -\frac{\partial H(P,X)}{\partial X} = F(P,X) \quad (3b)
\end{align*}
\]

IDEAL NON-STRETCHABLE TUGGERS

For the general case of tugger with \( V = V(P,X) \) it follows that

\[
dV = \left( \frac{\partial V}{\partial P} \right) \cdot dP + \left( \frac{\partial V}{\partial X} \right) \cdot dX \quad (4)
\]

so that increasing the PRESSURE P at fixed STROKE X will generally produce an increase of volume through stretch of the walls. But as discussed further below, it is usually desirable to design and construct tugger with nearly non-stretchable shells or skins. This is definitely the case with the cylinder-walls of fluid rams for obvious reasons. Moreover, unstable ballooning-action of the tugger walls must be avoided to prevent blow-out ruptures. We may then impose this condition in the form:

\[
\frac{\partial V}{\partial P} = 0; \quad \therefore \quad V = V(X) \quad (5)
\]

For this condition to hold, the ENTHALPY must necessarily be of the form:

\[
H(P,X) = -P \cdot V(X) \quad (6)
\]

EXTENDED BICONE MODEL

In an effort to obtain a much-simplified thermodynamic expression to characterize AMPFLEX and DYNAFLEX tugger behavior, the writer in 1971 made use of the simplified EXTENDED BICONE model of Figure 7.

If one assumes non-stretchable geometry with \( V = V(X) \), this results in the expression for ENTHALPY:

\[
H = P \cdot \pi \cdot \left( D^2 - Y^2 \right) \cdot \left( 2Y/3 + M \right) \quad (7a)
\]

so that:

\[
\text{VOLUME} \ V = \pi \cdot \left( D^2 - Y^2 \right) \cdot \left( 2Y/3 + M \right) \quad (7b) \\
\text{STROKE} \ X = 2 \cdot (D - Y) \quad (7c)
\]

We may then directly compute the relation between FORCE F and Y, so as to obtain the ratio of FORCE to PRESSURE as an EFFECTIVE AREA, \( Ap \), where

\[
Ap = F/P = \pi \cdot \left[ Y \cdot (Y+M) - D^2/3 \right]. \quad (8)
\]

This quantity is precisely equivalent to the actual BORE AREA of a fluid ram. Thus it is useful to express this stroke-dependent AREA in a purely dimensionless form as follows:

\[
a = Ap / (\pi D^2/6) = 1.5(y + m) - 0.5 \quad (9)
\]

where we have set \( y = Y/D \) and \( m = M/D \).

For the special case of the IDEAL BICONE, where \( m = 0 \), the PERUNIT AREA \( a \) reduces to

\[
a = 1.5 \, y^2 - 0.5. \quad (10)
\]

These results can, of course, also be expressed in terms of a suitable dimensionless STROKE, \( x \), as treated further below.

GAYLORD CROSS-BRAID DESIGN

The Bridgestone RUBBERTUATOR is based upon certain significant improvements made on the once-patented CROSS-BRAID design of Richard Gaylord (1958). Particularly, steps were taken to minimize the abrasion and friction due to the motion of the braids across each other and over the inner tube. Gaylord had previously established simple equations for both the tensile FORCE, \( F \), and the pressurized contracted LENGTH, \( Z \), of the tugger. In dimensionless form, corresponding roughly to the BICONE model, these respectively become

\[
a = f/p = 3 \, \cos^2 \theta - 1 \quad (11a) \\
z = \cos \theta \quad (11b)
\]

or, upon eliminating the braid angle, \( \theta \), there results

\[
a = 3 \, z^2 - 1 \quad (12)
\]

But because the relatively long RUBBERTUATOR could benefit from a filler-body to minimize air-consumption by the non-working dead-volume within the tugger, Bridgestone researchers also computed the gross interior volume corresponding exactly to the Gaylord results above, to obtain, again in dimensionless form

\[
v = \sin^2 \theta \cos \theta = \cos \theta \cdot \cos^3 \theta \quad (13)
\]

or, eliminating \( \theta \), \( v = z - z^3 \). Because this analysis assumes no stretch, one can immediately determine the PERUNIT AREA, \( a \), namely \( a = -\partial v / \partial z = 3 \, z^2 - 1 \), which, besides being identical to the result above, agrees precisely with the IDEAL BICONE result!

\[
a = 1 - x - (2 + k - \sqrt{3})(1 - x) \quad (15)
\]

It is this last form which we call the PANTOGRAPH EOS, with the appearance of Figure 8, whose two curves correspond to \( k = 0 \) and \( k = 0.5 \). Finally, if the results of the EXTENDED BICONE model are similarly plotted, they are indistinguishable from the above EOS using small values of \( k \).

MEASURED TUGGER CHARACTERISTICS

The measured FORCE vs STROKE characteristics of a large, low-pressure AMPFLEX tugger are presented in Figure 9, as determined in 1969. A direct display of the actual tugger stretch can then be found by plotting, as in Figure 10, the measured \( Ap = F/P \)
against tugger contractile STROKE, X. Using appropriate parameter values, D and M, in the BICONE model, corresponding to each PRESSURE, P, then yields a direct estimate of the tugger skin-stretch. It is clearly quite small for this particular tugger.

Similar FORCE-STROKE characteristics for a small, high pressure DYNAFLEX Model 125 tugger are given in Figure 11. These can then be reduced to a x vs x plots as indicated in Figure 12. The result is essentially fitted by a single curve, closely approximated by a PANTAGRAPH EOS with k = 0.5 as was depicted in Figure 8.

Finally, presented in Figure 13 are similar characteristics for a high-pressure Bridgestone Model No. 15 RUBBERTUATOR, comparable to those indicated in Figures 2 and 3. The indicated hysteresis arises in slewing this tugger over its full-stroke range for each pressure. It is also interesting that Bridgestone uses this same PANTAGRAPH EOS to calibrate their measured characteristics.

CONCLUSION

Not only does this thermodynamic treatment inter-relate all existing tugger designs, but it directly suggests other significant improvements not only in tugger fabrications (Paynter, 1988a,b) but also, by establishing an analogous ANGULAR ENTHALPY H(P,α), one may conceive and construct a new class of twistors (Paynter, 1978). These have been discussed in a companion paper and are treated thermodynamically in Part 2 of this present paper.

REFERENCES


Figure 1... Arthrobot

Figure 2... Bridgestone SoftArm

(a)... Ampflex

(b)... Dynaflex

(c)... Rubbertuator

Figure 3... Three TUGGERS
Figure 7... BICONE Model

Figure 8... Gaylord Pantagraph EOS

Figure 9... Ampflex Characteristics

Figure 10... Ampflex $A_p$ vs Stroke
Dynacyle Corporation
Dynaflex Model D125 Characteristics

Figure 11... Dynaflex Characteristics

Figure 12... Dynaflex a vs x

Bridgestone Corporation
RUBBERTUATOR
Characteristics
Size # 15

Figure 13... Rubbertuator Characteristics
THERMODYNAMIC TREATMENT of TUG-
&-TWIST TECHNOLOGY Part 2: Thermodynamic Twistor Design

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Abstract—Various jointed-arm or jointed-leg robots called Arthrobots can use the new technical field variously called Rubberronics or Pneumo-elastics and designated here as Tug-&-Twist technology. Part 1 of this paper was concerned only with Linear Tension Actuators or Tuggers [1]. A thermodynamic treatment in Part 1 inter-related all existing Tugger designs through an Enthalpy relation \( H(P, \alpha) \). Now by establishing an analogous angular Enthalpy \( H(P, \alpha) \), one may conceive and construct a new class of Twisters. Part 2 of this paper introduces a transforming transducer, in the form of a fluid-driven torsional device called a Twistor actuator. Analysis for this device is based upon the application of thermodynamic principles and in particular, the use of this Enthalpy function. Volume measurements then show that the thermodynamic treatment leads to an approximate model that predicts output torque taking into account the effect of shell stretch due to pressure.

Index Terms—Arthrobots, Twistor, Legendre transformation

I. INTRODUCTION

This paper, as Part 2 of a thermodynamic treatment of Tug&Twist Technology, is here concerned with Twistor actuators [1], [2], [3], [5], [6], [7]. Configurations of robots wherein all degrees-of-freedom are joint rotations has meant that torque motors could be used at each and every axis [4]; such devices have been designated Arthrobots [1], [2], [5], [6]. A Twistor is a Pneumo-elastic device that is an inflatable actuator having an axially elongated, flexible, hollow, thin-walled elastic cylindrical shell defining a fluid chamber, Figure 1, [2], [3], [5], [6], [7]. The Twistor device shown in Figure 1 is pretwisted 180 degrees into the jointed actuator shown in Figure 2. By controllably varying fluid pressure fed into the elastic shell when the torsional actuator is inflated, its flexible shell expands, causing the pretwisted actuator to uncoil and rotate the torsion bar in Figure 2, with \( P_1 > P_2 \).

This paper partially analyzes performance of a Twistor actuator by experimentally measuring the volume at various pressures while maintaining a constant set of angles, \( \alpha \) as shown in Figure 2 and through the thermodynamic

Figure 1
II. THERMODYNAMIC APPROACH TO DESIGN

Our approach is based directly upon the First Law of Thermodynamics, recognizing the near-reversible conversion of fluid available energy into mechanical energy. One of the authors has employed such methods for Twistor design during the last 25 years and more recently suggested their use for twistors [1].

This approach begins with the First Law statement:
\[
de = P \cdot dV - T \cdot d\alpha
\]
where Energy \( E = E \, (V, \alpha) \)

This represents a constraint between Pressure, \( P \); Volume, \( V \); Torque, \( T \); and Twist Angle, \( \alpha \). However, here we are interested in the situation where \( P \) and \( \alpha \) are proper Input variables with \( V \) and \( T \) being the corresponding Output variables [6], [8], [9], [10], [11], [12]. To create these conditions, we must perform a Legendre transformation upon Energy to form a complementary energy or generalized Enthalpy, \( H \) [13], [14], [15].

The following Legendre transformation may be performed as follows using the same procedure in [1] which is identical to the Legendre transformation, as in, say, Goldstein [16].

Given \( E(V, \alpha) \), the function \( H(P, \alpha) \) is found from the thermodynamic relation:

\[
H(P, \alpha) = E - P \cdot V \\
dH = -V \cdot dP - T \cdot d\alpha
\]

From this relation we then obtain the Outputs:

Volume, \( V(P, \alpha) = - \frac{\partial H(P, \alpha)}{\partial P} \)
Torque, \( T(P, \alpha) = - \frac{\partial H(P, \alpha)}{\partial \alpha} \)

These results suggest that the active transduction torque may be estimated by integration of the volume changes with pressure.

III. EXPERIMENTAL VOLUMETRIC RESULTS

Volume data was collected on a single Twistor actuator. A Twistor in the untwisted position is said to be at zero degrees when the chamber is fully pressurized. The data that was collected resulted in volumetric measurements taken at 50, 60, 70, and 80 degrees of Twist Angle; and various pressures of 35, 50, 60, and 70 psi. Measurements were made at 80 psi but, at this point the actuator began to balloon excessively as was predicted by the designer, recognizing that increasing the shell wall thickness would support additional pressure but would detract from low-pressure response. Table 1. below shows the results of the experiment.

<table>
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<th>( P ) / ( \alpha )</th>
<th>50</th>
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<th>70</th>
<th>80</th>
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<td>3.03</td>
<td>2.78</td>
<td>2.42</td>
<td>2.25</td>
</tr>
</tbody>
</table>

Table 1. Column 1 represents Pressure settings in pounds per square inch, Row 1 represents Twist Angle settings in degrees, and the interior points represent the measured Volumes in cubic centimeters.

The volume data was fit to an equation that maintains the Maxwell Relations and the differential form of the First Law equation 1, [17], [18].

An equation for volume found by using Microsoft Excel linear regression after converting the experimental quantities to SI units is:

\[
V(P, \alpha) = 1.67 \times 10^{-6} + 2.97 \times 10^{-12} \cdot P - 5.11 \times 10^{-9} \cdot \alpha - 7.7 \times 10^{-13} \cdot \alpha \cdot P
\]

Where \( V \) is in \( M^3 \), \( P \) is in Pascals, and \( \alpha \) is in Radians.

Figure 3. Linear Regression Volume, \( V(P, \alpha) \) below shows the equation's fit through the volume data as function of twist angle at the four experimental pressures. \( V(P, \alpha) \) is within one standard deviation from the experimental data results in Table 1.
The Enthalpy, \( H(P, \alpha) = \int V(P, \alpha) \, dP \) which follows upon integration, yields:

\[
H(P, \alpha) = 1.67 \times 10^{-6} P + 2.97 \times 10^{-12} P^2/2 - 5.11 \times 10^{-6} \alpha P - 7.7 \times 10^{-13} \alpha P^2/2
\]

And, \( T(P, \alpha) = -\partial H / \partial \alpha = 5.11 \times 10^{-6} P + 7.7 \times 10^{-13} P^2/2 \)

The active torque in Figure 4 below is finally computed by substituting pressure into the isobaric torque equation above to get constant torque lines for pressures ranging from 100kPa to 500kPa. Torque for this small actuator is in Newton-Meters.

### IV. CONCLUSION

This analytical approach is very effective for estimating active torque when only volume data is available for twistors. It has been shown that the volume equation in the form above satisfies the Maxwell relations and that the intermediate evaluation of the (partial) Enthalpy serves as the means toward calculating active Torque.

Although it is clear from the above test results that this particular twistor has only a limited torque capability, nevertheless scaling-laws readily extrapolate these test characteristics to larger sizes. For example, suppose we wish to provide for torques some 100 times greater than those obtained above. The thermodynamic analysis tells us that the expression \([\text{pressure}] \times [\text{volume}] / [\text{angle}]\) must itself be 100 times greater.

- While gearing-down by a factor of 1/100 would indeed furnish the desired torque it would result in an impractical angle-variation of less than a half-degree.
- Thus, if the angular swing is to be maintained at the test range, the \([\text{pressure}] \times [\text{volume}]\) product must itself be 100 times greater. Because increasing the pressure 100-fold is unreasonable, the net volume must instead be increased by this factor. While this could be accomplished by an array of 100 twistors of the size tested, it is more practical to use a single twistor having linear dimensions only 4.64 times larger than the tested device, where the factor 4.64 is simply the cube-root of 100.

### V. ACKNOWLEDGEMENTS

A special thanks goes to Mario DiMarco of DM3 Electrics for use of his experimental facility and for suggesting that we use air over water to facilitate measuring the actuator chamber volumes.

### VI. REFERENCES


