INNOVATIONS IN SMALL EXTRUDERS
THAT
PROMOTE FEEDING AND PRESSURE STABILITY

BY

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Abstract: Historically, small extruders, (defined here as about one inch or smaller screw diameters) had notorious feeding problems. These feeding problems have, in turn, caused surges. This paper describes innovations that change the feeding characteristics of small screws and a Surge Suppression Device.

History: Most feed stocks are pelletized. Most commonly, they take the form of spheroids, and cylinders though they may be cubes or hexagonal. Typical pellets have are nominally 0.13 inch (1/8 inch) but many pellets have a major dimension about 50% larger or about 0.19 inch (3/16 inch). The majority of extruded plastics are processed through large extruders. However, in the medical industry the cross sections of products such as catheters are so small that extruders must be correspondingly small. Large extruders that run very slowly suffer from long residence time and subsequent polymer degradation.

Small extruders, on the other hand, can process plastics at conventional screw speeds thus avoiding degradation. However, as extruder screws get smaller, conventional pellets become relatively larger. Over time, several problems were identified:

1) Packing Density: Pellets must fit in the feed channel of the extruder screw. If you design a one inch screw with a feed channel depth of about 0.19, most pellets will fit into the channel and feed. Pellets can then pack side by side in a single layer. Contrast this with the packing that can happen in larger extruders where pellets begin to pack three dimensionally somewhat like cannon balls. This higher density packing in larger extruders is advantageous because the feed channel will more likely be regularly filled. Uniform filling in the feed channel promotes uniform pressures. Without this uniform packing, small extruders tend to have an erratic feed and consequently less stable melt pressure.

2) Weak Screws: The obvious solution to the problem of a small channel depth is to make a larger channel depth. One inch screws can be made with channel depths up to about 0.24 inches in feed channel depth. However, such larger channel depths weaken the screw very substantially. Such screws are easily broken in the solids conveying zone because the load is bigger than the screw root can withstand.

A second problem has also been observed. Small dies, rather common with small extruders, often generate substantial pressures of between 3,000 and 7,500 psi. This pressure pushes on the tip of the screw. One inch screws with feed channel depths in this range can fail from this pressure. That is, these screw have been observed to "compress" making the root diameter grow and compressing the screw pitch.

Both problems were even worse for extruders smaller than one inch. Three-quarter inch extruders, for example, could not practically be built with channel depths larger than 0.190 inches. Even so, such screws had significant feeding and screw breakage problems.

Screws smaller than three-quarters of an inch were generally thought impractical because they did not survive when tried.

3) Feed Throat Design: Conventional extruders have a hole in the barrel where the pellets fall by means of gravity into the screw. Usually, a separate water cooled section of the barrel is designed to prevent polymer from melting prematurely causing a lack of feed. This section of the barrel is called the "feed throat" or "barrel feed section." Large extruders pass conventional pellets readily through the feed throat to the screw channel. While several feed throat designs are possible, larger extruders are often fed from the top through a hole smaller than the screw diameter. The literature describes different types of smooth bore feed sections. Among these are a top dead center feed; a tangential design where the feed is offset from the screw diameter but vertical; and a tangential design where one side of the feed is angled thus forming a wedge with the feed. The tangential designs are recommended for melt fed rather than solid feed stocks. Another type of smooth bore design is known for the rubber industry to as a roll feeder and is designer to feed in strips of material rather than for typical pelletized feed stocks.

Several texts sketch the dimensions of the feed throat. It appears from the scale of such drawings, that the barrel holes are somewhat smaller than the screw barrel diameter across the screw and about the same length as the barrel diameter along the screw axis. It is interesting that the design of the feed throat is given so little attention as it implies that the dimensions of the feed opening do not matter very much.

Manufactures of small extruders have long known that the size of the feed throat matters greatly. Typical pellets will readily "arch" over a diminutive three-quarter inch opening in a one inch extruder. This "arching" stops material from reaching the screw. Consequently, the feed sections have been "enlarged" by most manufacturers. One manufacturer, for example, enlarges the feed throat opening to the diameter of the screw (across the screw) and to two times the screw diameter in the axial dimension for their one inch and three-quarter inch extruders. To a significant degree, this solves the arching problem on the one inch size extruder but the effect is lessened on smaller sized extruders.

While arching is reduced, a consequence of the increased feed opening is a reduction in solids conveying. This is because the large hole lessens the barrel contact with the pellets which in turn *reduces solids' transportation*. Another consequence is that the larger feed opening comes at the expense of uniform water cooling and at the expense of the feed section's L/D ratio. This creates feed throats with temperatures that may be about 60F at the six o'clock position and 250F at 12 o'clock because of a lack of water cooling in this area. Such designs lack reliable solids conveying because radial temperature regulation is so poor.

The root causes of these problems were pretty much ignored by small extruder manufacturers and treated as insurmountable. Instead of addressing these problems directly, they offered three "solutions" to these problems:

1) Grooved Barrels (also called "Grooved Feed Throats and Grooved Feed Sections"): To solve the problems of inferior feeding, grooves were added to the feed sections in the 1980's. Grooved feed throats have one or more grooves in their bore. Usually, these grooves are parallel to the axis of the screw and are rectangular but they may be hemispherical, trapezoidal, and helical. The grooves effectively trap the pelletized feed stocks in the grooves against the screw helix increasing the coefficient of friction by about two or three times. Consequently, transportation increases substantially and the screw design is altered accordingly. Typical compression ratios are decreased from about 3:1 to about 1:1.

Several variables are known to contribute to feeding in grooved barrel feed sections. The number of grooves, the length of the grooves, and the shape of the grooves can all be tailored to specific materials. Thus, the machine designer and processor have a range of choices in grooved barrels to meet his requirements.

Several manufactures offer both smooth and grooved bore barrels. Interestingly, smooth bore extruders have remained more popular in the United States than grooved bore barrels even though grooved bore barrels offer significant advantages in many respects. Possibly, this is because smooth bore feed sections are more flexible than grooved bore barrels. That is, grooved bore barrels are designed for specific materials and may not allow for a very wide range of polymer processing (unless expensive feeders are used).

For both smooth bore designs and grooved bore designs, typical horizontal extruders place the feed section of the barrel between the main portion of the barrel and the thrust section. This natural placement makes it difficult to change from one type of feed section to another. To make a change on a one inch extruder, the screw must be removed. This might take 15 minutes to 1 hour depending on the material. The barrel cover must be removed and the screws that hold the barrel must be removed. There are two other considerations in that the barrel may be hot (from the heat required to remove the screw) and the barrel wires might have to be disconnected. This may also be time consuming and may involve additional people in the process. The screws that hold the feed section to the barrel are then removed. The feed section is replaced and the extruder is reassembled. So, the replacement process is somewhat time consuming even on a small typical extruder and this makes for delays in production.

The screw used with a grooved throat must still allow the pellets to fit into the screw channel. So, the one inch screw is usually equipped with a feed channel depth of at least 0.180. Since the "metering depth" is the same as this feed depth, the output of the screw is about two to three times higher for the same screw speed. It should be noted that high output is counterproductive in the manufacture of small cross sectional products such as catheters. So, while grooves increase the solids conveying and yield substantially more uniform pressures, they do so at a cost of higher output.

2) Gear Pumps: Gear pumps are well known to yield very stable pressures and under some circumstances seem the best way to achieve uniform outputs. They do have well know disadvantages including expense, complexity of operation, are not necessarily perfect "In/Out" pumps (so degradation is possible), and are tedious to dismantle and clean.

In any event, even when necessary and appropriate, gear pumps should not be used with poorly feeding extruders. This is because a gear pump only makes the output more uniform volumetrically. It does not improve the quality of a poorly melted or mixed extrudate that results from erratic solids' transportation (feeding).

3) Dual Diameter Screws: One company has recently displayed a dual diameter screw design rather similar to the "Pirelli Rubber Extruder." The soft rubber deforms in the conical feed throat where there is a large clearance between the screw and wall. This type of extruder has also found used in larger extruders where it is used to densify scrap such as the fluff made from ground bags. This type of feed stock is also soft in the sense that there is so much air in the feed stock (unlike dense hard pellets) that the feed stock is readily compressible.

The extruder displayed had a 3/4 inch feed section followed by a 1/2 inch screw. Neglecting the earlier comments about the strength of a 3/4 inch screw, uniform cooling requirements, feeding, and barrel feed geometry, it is worth noting the following:

a) Changing Screws: The barrel must be removed in order to remove the screw for cleaning or changing the screw.

b) Expensive Screws: It is very likely that dual diameter screws and barrels will be more expensive than screw of a single diameter when replaced.

c) Screw Design: It must be remembered that any screw is a balancing act. The solids conveying zone must transport the correct amount of material to fill the metering section of the section. The 3/4 inch screw should have a feed channel depth large enough for typical pellets. It is likely that the second screw diameter will have a rather large thread depth to accommodate the relatively large volume of material from the larger 3/4 inch screw. It may be difficult to balance the feed amount with the metering.

d) Wear: The exhibited extruder had a relatively short transition between the first and second screw. Unless rather slow screw speeds were used, one might expect this sudden transition to be a significant wearing zone for the barrel and screw as conventional hard pellets (compared to the traditional soft rubber and soft fluff applications where such extruders are more conventionally used) deform in the diminishing space of the tapered barrel.

INNOVATIONS

I) INTRODUCTION: This paper describes three innovations that yield more stable pressure and consequently more uniform products. The first innovation was the discharge driven extruder that Randcastle commercialized in 1988. In turn, this lead to two more innovations that can give more stable pressures. We will describe the behavior of different smooth bore feed throats and a patent pending Surge Suppression Device.

II) DISCHARGE DRIVEN EXTRUDER DESIGN: This first innovation was the introduction of a vertical extruder driven from the discharge end of the extruder. This design solves some of the historical problems that were generally thought insurmountable. As has been discussed, screw strength was a limiting factor that had stopped machine builders from making extruders smaller than about three-quarters of an inch.

In a typical extruder, the entire load of the extruder is transmitted through the root diameter of the screw under the hopper. This is usually where the root diameter of the screw is smallest and consequently weakest.

The formulae for allowable stress for main power-transmitting shafts (using 4,000 pounds per square inch) can be used as a simple approximation of the screw root diameter:

$$P = \frac{D^3 N}{80}$$

where: P= Power transmitted in horsepower
D= Diameter of the shaft in inches

N= Angular velocity of shaft in revolutions per minute
Using a three-quarter inch diameter screw having a channel depth of 0.180 inches as an example, the root diameter of the solid conveying region would be about 0.390 inches. At 80 revolutions per minute, 0.390 x 0.030 x 0.390= 0.059.

In a discharge driven design, the entire load of the extruder is transmitted through the metering section root diameter. Typically, this root diameter is significantly bigger. Using a typical 3:1 apparent compression ratio for the meter channel depth and the same feed channel depth of 0.180 inches, the meter channel depth would be 0.060 inches. The root diameter for the meter would then be about 0.63 inches. At 80 revolutions per minute, 0.63^3 equals 0.25.

Dividing, 0.25/0.059 = 4.23. So, the same screw driven from the discharge end of the screw is about four times stronger than a conventional screw. This is not completely correct of course since some of the load is transmitted through the tapering root diameter of the melting zone. There is every reason to believe, however, that discharge driven screws are substantially stronger than conventionally driven screws. In practice, discharge driven extruders are now built as small as 0.25 inch in screw diameter and 0.500 inch diameter for pelletized feedstocks.

III) FEED THROATS FOR DISCHARGE DRIVEN DESIGNS:

A) Arching Resolved: As discussed earlier, one of the problems with conventional small extruders was getting the pellets to the screw. Arching (mechanical bridging of the pellets over the feed throat opening) was a significant problem. It caused poor feeding because the pellets did not arrive regularly at the screw channel. Once the extruder is discharge driven, the problem of getting the feed stock to the screw is resolved merely by extending the screw into the feed throat. See Drawing 1.

In addition, because the screw is rotating within the feed section, the end of the screw can be modified to stir materials with the hopper. This is useful with materials that are not free flowing such as sticky pellets that have a tendency to "funnel" or "rat-hole" well above the feed throat. See Drawing 1.

B) L/D Properly Cooled: The feed throat is made with a chambered cooling system that is three L/D's long. The chambered cooling forces material to flow from one chamber to the next to insure uniform cooling. Because the feed section has a working cooled length of 3 L/D's, feed throat friction reliably transports material in this portion of the solids conveying zone.

C) Innovative Smooth Bore Feed Throats: Randcastle has devised a means to change how much material is transported by means of different types of smooth bore feed throats. This changes the packing density making the feeding behavior more like larger extruders. Earlier in this paper, it was noted that the typical choice offered the purchaser was either a smooth bore feed throat or a grooved bore feed throat. This choice alters the feeding behavior radically. As a consequence, the screw's apparent compression ratio must also be changed radically. Smooth bore apparent compression ratios are typically between 2:1 and 4:1 for smooth bore feed throats and about 1:1 for grooved bore feed throats. The amount of feed is therefore balanced with the metering section of the screw.

However, Randcastle has discovered that this general principle may also be applied to smooth throat feed sections as well. That is, Randcastle has developed three smooth bore feed throats for pelletized feed stocks where transportation (feeding) is changed with each type of feed throat. The major difference is that feeding is altered in smaller amounts compared to the radical changes that take place in smooth verses grooved feed throats.

D) SET-UP FOR TESTING: Randcastle offers four types of smooth bore feed sections. One type is the roller feed section designed for strip feed and will not be discussed in the paper. The other three types were installed on a Randcastle 5/8 inch extruder so that different materials could be processed and the effects observed. Specifically, we were interested in the feeding characteristics of the different types of feed sections with an eye towards more stable polymers. We wanted to know if we could alter the feeding characteristics without changing the screw. This would allow the option of changing either the screw to affect more uniform flow or changing the feed sections.

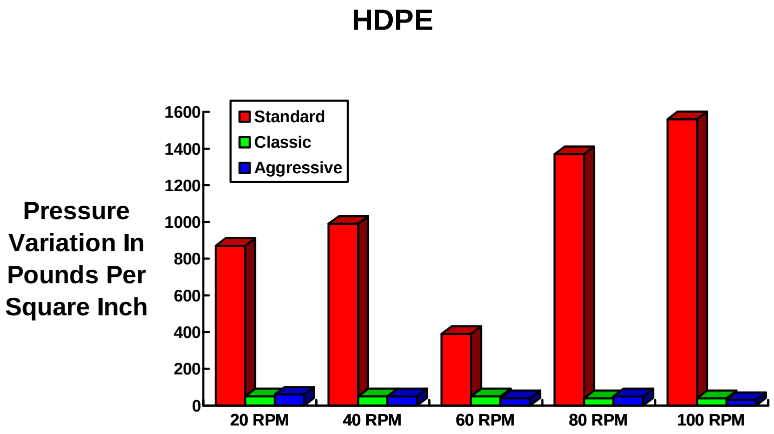
Unlike the conventional extruder, Randcastle's feed sections can be changed without removing the screw. Because the feed section is held in place with only four screws, the feed sections can be changed in about a minute. This means that production downtime can be minimized and catheter production increased.

The experiments were carried out using a single general purpose Randcastle screw design having a 3:1 compression ratio with 8 L/D's of meter, 8 L/D's of transition, and 11 L/D's of feed. The screw had a minimal impact Surge Suppressor installed to minimize short term pressure variation.

The specific smooth bore barrel feed section designs are, of course, proprietary. They will be referred to here as "Standard, Classic, and Aggressive." This is useful and necessary from an identification point of view. However, these are just names and the reader should not read too much into the names themselves.

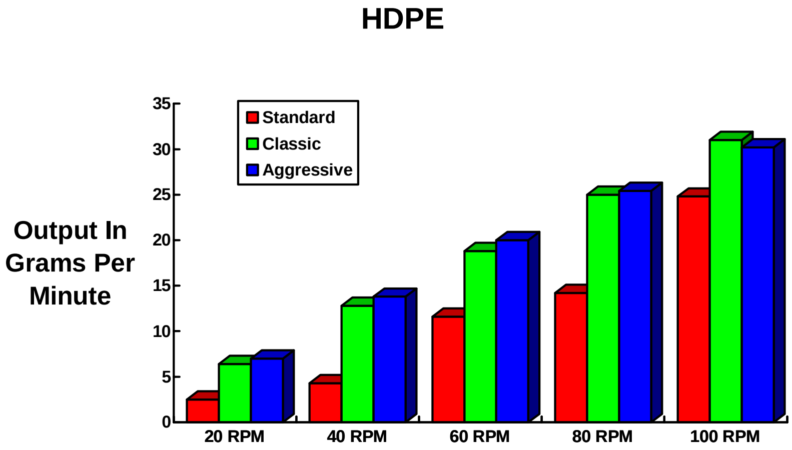
E) RESULTS OF FEED THROAT TESTING:

1) HDPE: The first material that was processed was HDPE from Federal Plastics #F15896. This was an underwater cut pellet. Barrel conditions for all trials were zone one 360F, Zone two 370F, Zone three 380 F, and the die at 380F. The extruder used was a standard Randcastle 24/1 working L/D 5/8 inch Microtruder. The graphical results are:



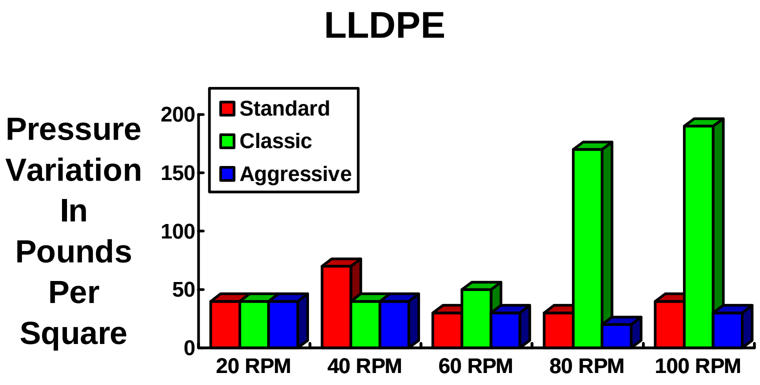
The conclusion is that the "Standard" feed section was not stable but that both the "Classic" and "Aggressive" smooth bore feed sections produced very good average fluctuations during the test. Averaged pressure fluctuations for the "Classic" feed section were plus or minus 23 psi and for the "Aggressive" feed section plus or minus 22 pounds. Average output for the "Classic" barrel feed section was 0.30 grams per revolution while the "Aggressive" feed section yielded 0.32 grams per revolution.

The output for the HDPE was:

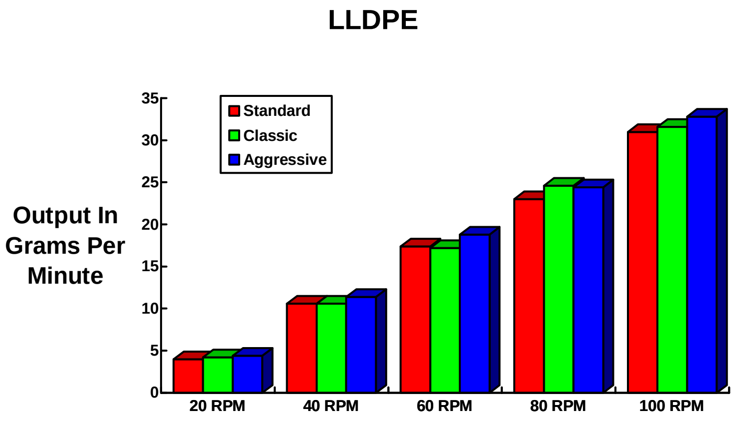


The output from both the "Classic" and "Aggressive" feed throats are much higher than the "Standard" feed throat and both produced stable pressures. This implies that the "Standard" feed throat supplied too little material to the metering section and it was consequently starved and surged.

2) LLDPE: The next material that was tested was LLDPE at barrel zone temperatures of 385, 390, 400, and 400 F from the feed to the die. The pressure variation follows:

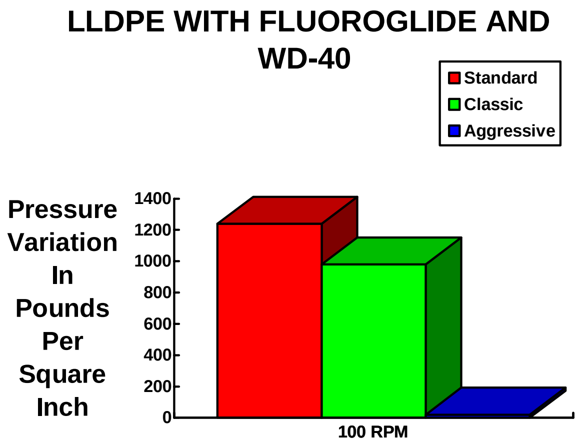


The output graph for the LLDPE follows:

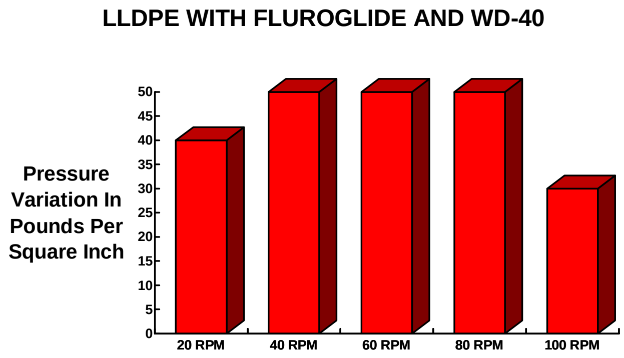


As in the case of the HDPE, the output is consistently higher when changing from the "Standard" to the "Classic" to the "Aggressive" feed throats. Unlike the HDPE trial, the output fluctuation for the "Classic" feed throat is probably not because the metering section is starved. After all, the average output values are lower for the "Standard" feed throat (compared to the "Classic" feed throat) but higher for the "Aggressive" feed throat. It seems more likely that some other aspect of the process is causing the instability.

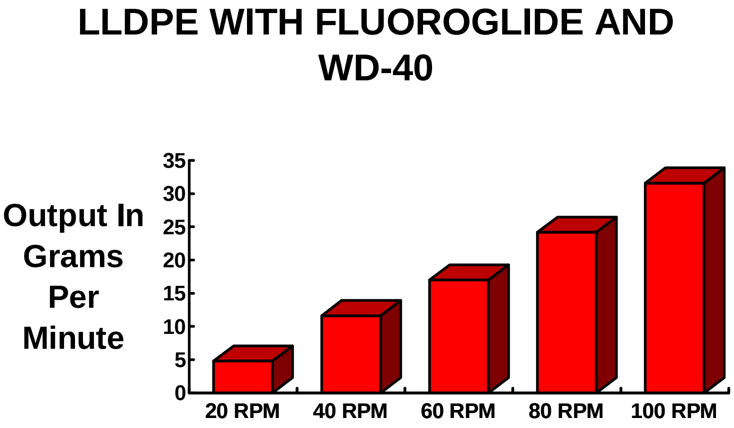
We then modified the LLDPE but modified to make it excessively slippery. To this, we sprayed the feed stock with an aerosol mixture of "Fluroglide" and "WD-40" and mixed the pellets to distribute the spray. Processing conditions were kept the same as during the virgin tests above. Under these circumstances, the "Standard" feed throat and "Classic" feed throat produced wildly unstable pressures while the "Aggressive" feed throat did not. The following graph shows the approximate fluctuations at 100 RPM for all three feed throats:



The pressure variation for the complete run on the LLDPE follows:

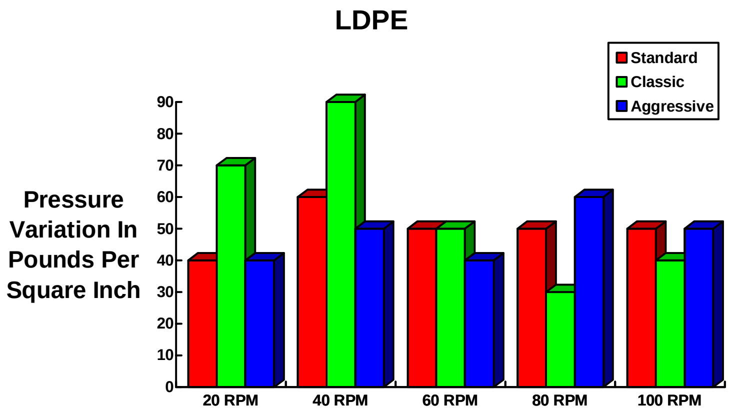


And the output for this trial was:

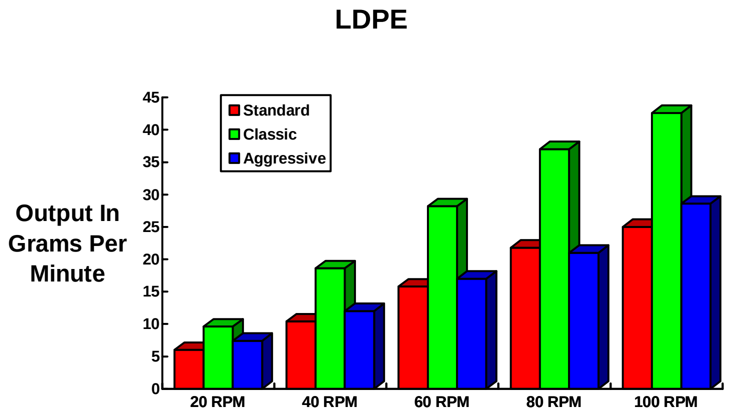


This is a rather interesting result. If you compare the output using the virgin LLDPE and the LLDPE with the Fluoroglide and WD-40 using the "Aggressive" feed throat, they are very similar. Since this pressure was so unstable with the "Standard" feed throat, the obvious conclusion is that the "Classic" feed throat is sensitive to changes in the feed stock's coefficient of friction that the "Aggressive" feed section is not sensitive to.

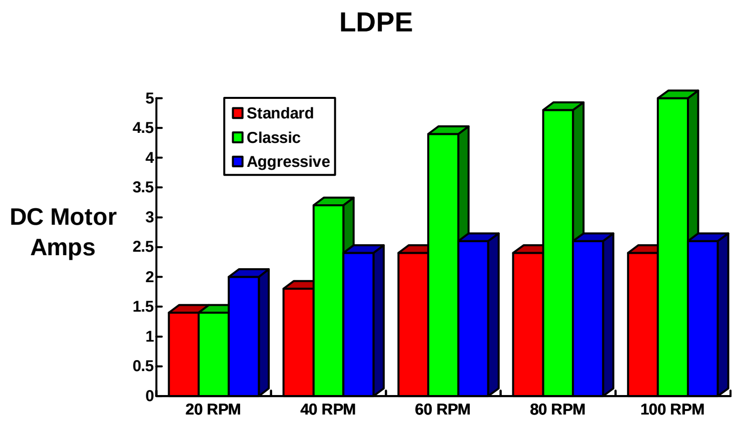
3) LDPE: The next material tested was a Federal LDPE #Nat:F13600 at temperatures starting at the hopper and moving progressively down the die from 300, 325, 350 and 350F. The output pressures were:



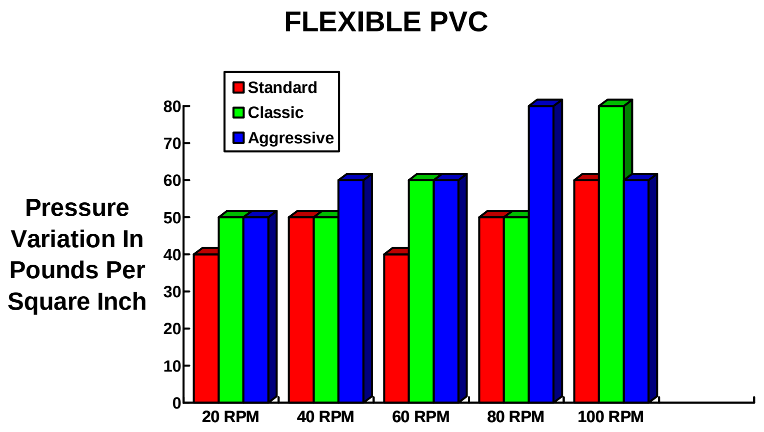
The pressure for the LDPE tested was:



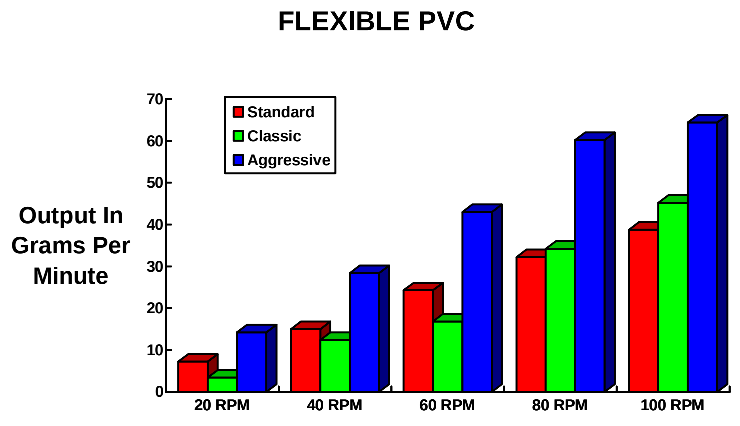
Apparently, in this case, the "Classic" feed section fed much better than either of the other two feed sections. Apparently, it fed too well and as a result overwhelmed (at this set of process conditions) the screw's metering section making the pressure unstable. Additional evidence may be seen in the motor amps shown below:



4) FLEXIBLE PVC: The last material tested was flexible PVC from Federal Plastics. It was a clear flexible underwater cut feed stock #F15763 and was processed with profile of 350 at the hopper, 345 at zone 2, 340 in zone 3 and 335 at the die. Pressure stability was:



In this case, the "Standard" feed section seems to have performed most reliably. The output was:



All these outputs seem strikingly high compared to the previous trials even given flexible PVC's high specific gravity.

F) CONCLUSIONS REGARDING THESE NOVEL FEED THROATS:

1) Effect Of Process Conditions: In these experiments, one particular set of conditions was selected for each material and for all the feed throats. This makes for good science but not necessarily for the most stable pressures. If process conditions were changed, the results might change. We made no attempt to find ideal (as measured by pressure) temperatures and, consequently, we doubt that we found them. We think that the general trends (like the "Standard" feed section forwarding the least material) will not be greatly influenced by typical processing changes.

2) Effect Of Screw Design: These experiments were all done with one screw. It is a rather ordinary design (3:1 apparent compression ratio where 8 L/D's were meter, 8 L/D's were compression, and 8 L/D's were feed). We expect different results with a different screw design. We base this expectation on our lab experience with a very similar screw different only in its 4:1 apparent compression ratio. If I summarized this study on its own, I might say that the "Aggressive" feed throat was, more often than not, the best choice; the "Standard" feed throat, more often than not, was the least useful. Our experience with the 4:1 screw is exactly the opposite: The "Standard" feed throat is most useful and the "Aggressive" the least. This does not surprise us. There is only so much room in the metering section of a screw. If you convey more material forward from the hopper because the screw's apparent compression ratio is higher, there is less room for material conveyed by an "Aggressive" feed throat.

3) Synergistic Effect: These results suggest something rather unexpected. Originally, we thought that we could use simply change these feed throats to convey more or less material per revolution as an aide to proper filling of the screw. We thought that this would simply be easier (because you can change feed throats in about one minute without removing the screw or die) to work with. We knew that feed throats cost less than screws so we figured this was good. But, we also thought that changing screws to another design would work just as well. Now, were not so sure. We think that, at this size extruder, the specific pellet shape, hardness, and friction interact with the specific feed throat. This interaction seems to cause a positive, negative, or neutral reaction in terms of pressure stability. Sometimes (looking at the "Classic" feed throat for LDPE) it seems to do both depending on screw speed. The question is, do these feed throat designs convey advantages beyond what a screw change might? We think so.

It is clear, for example, that the "Classic" feed section had a significantly higher output at all speeds for the LDPE trials. It is equally clear that the "Aggressive" feed throats had significantly higher output for the flexible PVC trials. Since the geometry of the feed throats did not change and since the pellets did not change, we must suppose that something else changes transportation. Similar logic might be applied to the trials LLDPE and modified LLDPE.

We think that pellet shape and hardness (interacting with the different feed throat geometries) are the most likely causes of these results. Pellet shape is probably important because of packing density. That is, spheres pack differently than cylinders or diced pellets. We think that these different feed throat geometries arrange or organize the pellets in different ways. Sometimes the reorganization yields more consistent packing and therefore feeding and more stable pressures. Sometimes not. We think that pellet hardness is probably involved too because hardness is related to shape change. And, when you are trying to get hard pellets to fit into channels just slightly bigger than the pellets, this becomes important.

IV: SURGE SUPPRESSOR DEVICE: Until recently, the only solution to improving pressure stability in an existing screw design was by means of fine tuning a particular screw design or by adding a gear pump to a screw. Through patent pending technology called, Surge Suppression, gear pump like pressures of plus or minus 10 PSI at 2,700 PSI pressure have been achieved on some small Randcastle Microtruders. See, *"Surge Suppression— A New Means To Limit Surging"* by Keith Luker.

ADDITIONAL SET UP CONSIDERATIONS

The HDPE, LLDPE, and PU trials were performed using a 0.076 rod die with a 5:1 land, and breaker plate without screens, and a Randcastle Model RCP-0625, 5/8 inch extruder with a 1 1/2 HP drive. The PU trials were conducted with limited amounts of material so it was not possible to do extensive tests. Long term drift is likely to be greater than the ~pressure recorded in the following charts.

All LDPE and flexible PVC trials were performed using a 0.060 monofilament die with a 10:1 land, a breaker plate with a 40, 80, 100 mesh screen pack and the same extruder.

MEAN PRESSURE: STANDARD FEED THROAT

HDPE	575	805	1375	1035	1120
LLDPE	960	1345	1595	1785	1960
LLDPE Modified	X	X	X	X	1260
LDPE	1760	2160	2535	2830	3115
FLEXIBLE PVC	1620	1755	1930	2045	2140
	20	40	60	80	100

RPM

MEAN PRESSURE: CLASSIC FEED THROAT

HDPE	1065	1365	1635	1830	1890
LLDPE	970	1330	1575	1735	1905
LLDPE Modified	X	X	X	X	1350
LDPE	2205	2415	3295	3595	3820
FLEXIBLE PVC	1460	1775	2010	2155	2240
	20	40	60	80	100

RPM

MEAN PRESSURE: AGGRESSIVE FEED THROAT

HDPE	1100	1435	1640	1825	1950
LLDPE	980	1380	1635	1850	1995
LLDPE Modified	925	1355	1585	1810	2010
LDPE	2000	2375	2600	2850	3085
FLEXIBLE PVC	1685	2150	2370	2560	2775
	20	40	60	80	100

RPM

TECOTHANE 55 D
STANDARD FEED SECTION

Run	Gr./min.	RPM	Zn.1	Zn. 2	Zn.. 3	Die	Melt	Pressure	Amps
1	15.2	20	400	420	430	430	431	71 to 82	3
2	25.4	40	415	420	420	400	431	170 to 180	2
3	35.6	60	415	420	420	400	440	240 to 260	4
4	37.2	80	415	420	420	400	440	330 to 350	5
5	42.4	100	415	420	420	400	440	480 to 490	5

TECOTHANE 55 D
CLASSIC FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	17.2	20	415	420	420	400	435	80 to 90	3
2	26.0	40	415	420	420	400	434	240 to 255	5
3	39.0	60	415	420	420	400	433	420 to 435	7
4	53.2	80	415	420	420	400	434	630 to 650	7.5
5	64.0	100	415	420	420	400	437	710 to 780	8

TECOTHANE 55 D
AGGRESSIVE FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	18.4	20	415	420	420	400	435	90 to 100	4.5
2	32.4	40	415	420	420	400	433	260 to 300	7.2

TECOTHANE 75 D
STANDARD FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	17.4	20	415	420	425	400	438	155 to 165	3
2	29.4	40	415	420	425	400	441	330 to 345	4.8
3	38.6	60	415	420	425	400	440	350 to 375	5.4
4	52.2	80	415	420	425	400	440	500 to 540	6.2
5	62.8	100	415	420	425	400	441	575 to 625	4.8

TECOTHANE 75 D
CLASSIC FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	17.0	20	415	420	425	400	436	160 to 200	3

2	30.2	40	415	420	425	400	438	400 to 500	5
3	42.6	60	415	420	425	400	438	740 to 760	6
4	53.0	80	415	420	425	400	439	940 to 960	6.6
5	63.4	100	415	420	425	400	441	1030 to 1060	7

TECOTHANE 75 D
AGGRESSIVE FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	18.8	20	415	420	425	400	437	85 to 95	3.8
2	32.2	40	415	420	425	400	438	205 to 215	5.8
3	42.2	60	415	420	425	400	438	330 to 375	7.2
4	56.4	80	415	420	425	400	438	625 to 660	7.6
5	-	100	415	420	425	400	439	-	Over 8

PELLATHANE 55 D
STANDARD FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	11.8	20	400	410	415	400	427	70 to 80	1.8- 2.4
2	20.6	40	400	410	415	400	430	90 to 100	3.4-4.0
3	30.2	60	400	410	415	400	437	130 to 175	3.8-4.6
4	40.8	80	400	410	415	400	439	300 to 340	5.0-5.6
5	50	100	400	410	415	400	439	625 to 650	6.0-6.6

PELLATHANE 55 D
CLASSIC FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	13.4	20	400	410	415	400	428	95 to 105	3.4
2	26.6	40	400	410	415	400	429	250 to 300	5.8
3	38.2	60	400	410	415	400	429	415 to 470	6.2-7.0
4	50.0	80	400	410	415	400	430	650 to 700	7.2-7.8
5	60.0	100	400	410	415	400	432	Not Taken	>8

PELLATHANE 55 D
AGGRESSIVE FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	8.2	20	400	410	415	400	428	240 to 250	2.0-2.6
2	18.4	40	400	410	415	400	414	750 to 800	4.2-5.0
3	30.6	60	400	410	415	400	418	1200 to 1250	4.6-5.0
4	42.4	80	400	410	415	400	425	1400 to 1550	5.8-6.0
5	50.2	100	400	410	415	400	432	1600 to 1680	5.6-7.4

PELLATHANE 75 D
STANDARD FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	17.2	20	410	420	425	410	424	500 to 650	4.0-4.6
2	28.0	40	410	420	425	410	426	840 to 885	6.2-6.6
3	38.6	60	410	420	425	410	425	890 to 950	7.0-8.0
4	48.2	80	410	420	425	410	442		7.6-8.0
5	61.6	100	410	420	425	410	443		7.2

PELLATHANE 75 D
CLASSIC FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	14.6	20	410	420	425	410	438	380 to 410	3.0-4.0
2	27.4	40	410	420	425	410	438	440 to 580	6.0
3	40.2	60	410	420	425	410	438	820 to 865	7.2-7.4
4	52.2	80	410	420	425	410	439	1180 to 1230	7.8-8.0
5	65.2	100	410	420	425	410	441	1260 to 1320	8.2-8.4

PELLATHANE 75 D
AGGRESSIVE FEED SECTION

Run	Gr./min.	RPM	ZN.1	Zn. 2	Zn.. 3	Die	Melt	~Pressure	Amps
1	16.4	20	410	420	425	410	434	395 to 410	4.3
2	32.2	40	410	420	425	410	437	545-	8

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