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GAS FLOWS SUPPORTING UMBILICAL DIVING – REQUIREMENTS AND MEASUREMENTS

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This report compiles information from various publications and letters about the MK 3 LWDS umbilical-supplied breathing system. New analyses were also conducted with various computational methods to describe the system and the gas supplies delivered to three divers at the maximum depth of the system, 190 fsw. One calculation method for steady state gas flow was based on equations for airflow in pipes. Electrical models were used to describe steady state flow with no divers or dynamic flow with divers. The models were solved by the PSPICE differential equation solver. All methods consistently illustrated that flow restriction occurs in 600-foot umbilicals and limits the emergency gas supply available at 190 fsw. Furthermore, using a 135-psi overbottom console pressure severely limits the ability of a diver on steady flow to simultaneously breathe and keep water out of his diving helmet.

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ABBREVIATIONS, ACRONYMS, AND SYMBOLS

acfm	Actual cubic feet per minute gas flow, not corrected to standard temperature. See also "scfm".
ANU	Authorized for Navy Use (NAVSEAINST 10560.2 series)
ambient	normal laboratory temperature (75 ± 10 °F)
I.D.	internal diameter
atm abs	pressure in units of atmospheres absolute
bpm	breaths per minute
chance	A colloquial expression for probability. A 1 in 5 chance is equivalent to a probability of 1/5 or 0.20.
cmH_2O	centimeters of water (pressure)
cu ft	cubic feet
°C	degree Celsius
°F	degree Fahrenheit
DSI	Diving Systems International
EDF	Experimental Diving Facility (NEDU unmanned test facility)
fsw	feet of seawater
ft	foot
I.D.	internal diameter
L	liter
J/L	joules per liter (unit respiratory work), equal to 1 kPa
kPa	kilopascals, unit of pressure; kilonewtons/meter, also equal to J/L^2
kts	knots, nautical miles per hour
LFE	laminar flow element
L/min	liters per minute (flow rate)
m	meters
mg/L	milligrams per liter (water vapor content)
msw	meters of seawater
NAVSEA	Naval Sea Systems Command

NEDU	Navy Experimental Diving Unit
O/B	overbottom pressure
ΔP	pressure differential, a measure of the magnitude difference between peak inspiratory and expiratory pressures
ppm	parts per million
$\overline{P}_v$	volume-averaged pressure or "resistive effort", formerly "work of breathing"
PV	pressure volume (pressure volume loop)
psia	pounds per square inch absolute
psid	pounds per square inch differential
psig	pounds per square inch gauge
RMV	respiratory minute volume in liters per minute
RE	resistive efforts (formerly "work of breathing")
Re	Reynolds number, a dimensionless ratio indicating the relative significance of viscous compared to inertial properties of fluid flow.
scfm	standard cubic feet per minute, volumetric flow corrected to standard temperature and pressure
SG	specific gravity
STPD	standard temperature and pressure (0 °C, 760 mmHg), dry
V_T	tidal volume in liters of air breathed in and out of the lungs during normal respiration
UBA	underwater breathing apparatus
USN	United States Navy

CONVERSION TABLE

To Convert From	To	Multiply By
kgf m/L	joules per liter (J/L)	9.807
psi	kilopascals (kPa)	6.895
feet	meters (m)	0.305
fsw	msw	0.307
fsw	bar	0.031
fsw	kilopascals (kPa)	3.063
cmH_2O	kilopascals (kPa)	0.098
inch	centimeter (cm)	2.54
gallons	liters	3.79
psi	bar	0.06895

INTRODUCTION

The issue of maximal, minimal, and allowable gas flow rates for gas systems such as the Fly-Away Dive System (FADS) and MK 3 Light-Weight Dive System (LWDS), supporting MK 20 and MK 21 diving has been discussed and reviewed at both the Navy Experimental Diving Unit (NEDU) and NAVSEA 00C for at least the past 10 years. Appendix A lists the pertinent publications and some of their conclusions. This report seeks to compile those publications and add new material to further explain and supplement the earlier work.

Recently, there has been some concern that the flow estimates published in NEDU's 1995 report[1] on the MK 3 LWDS may have been too high. One of the purposes of this analysis was to confirm from first principles whether or not the reported flow rates were reasonable.

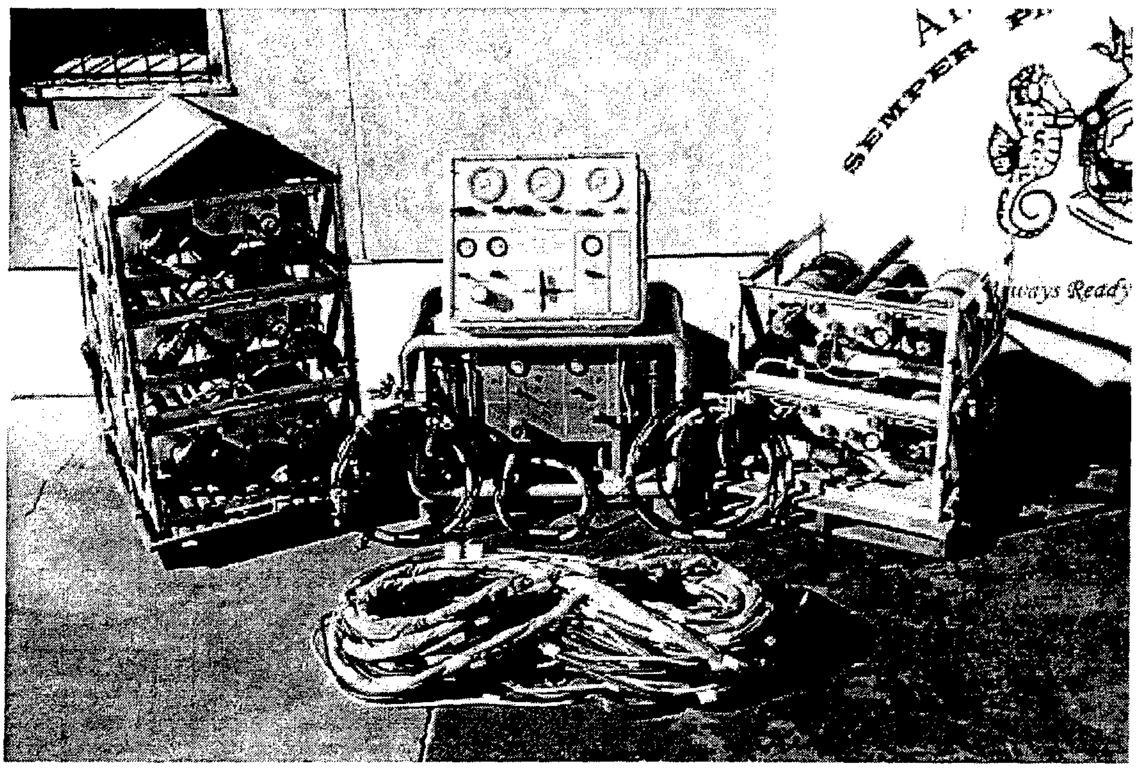

Figure 1. MK 3 Light Weight Dive System. (U.S. Navy Diving Manual, Rev. 4.)

This report is divided into two sections. Section I describes the procedure whereby, accounting for console and umbilical pressure losses, we calculated flow rates available to a MK 21 diver for a given MK 3 LWDS console pressure. It also uses the convention of electrical analogs to simulate respiratory and umbilical flows.

Section II provides information on potential flow measurement devices that could be used in the field to check the flow output of both MK 3 and FADS consoles.

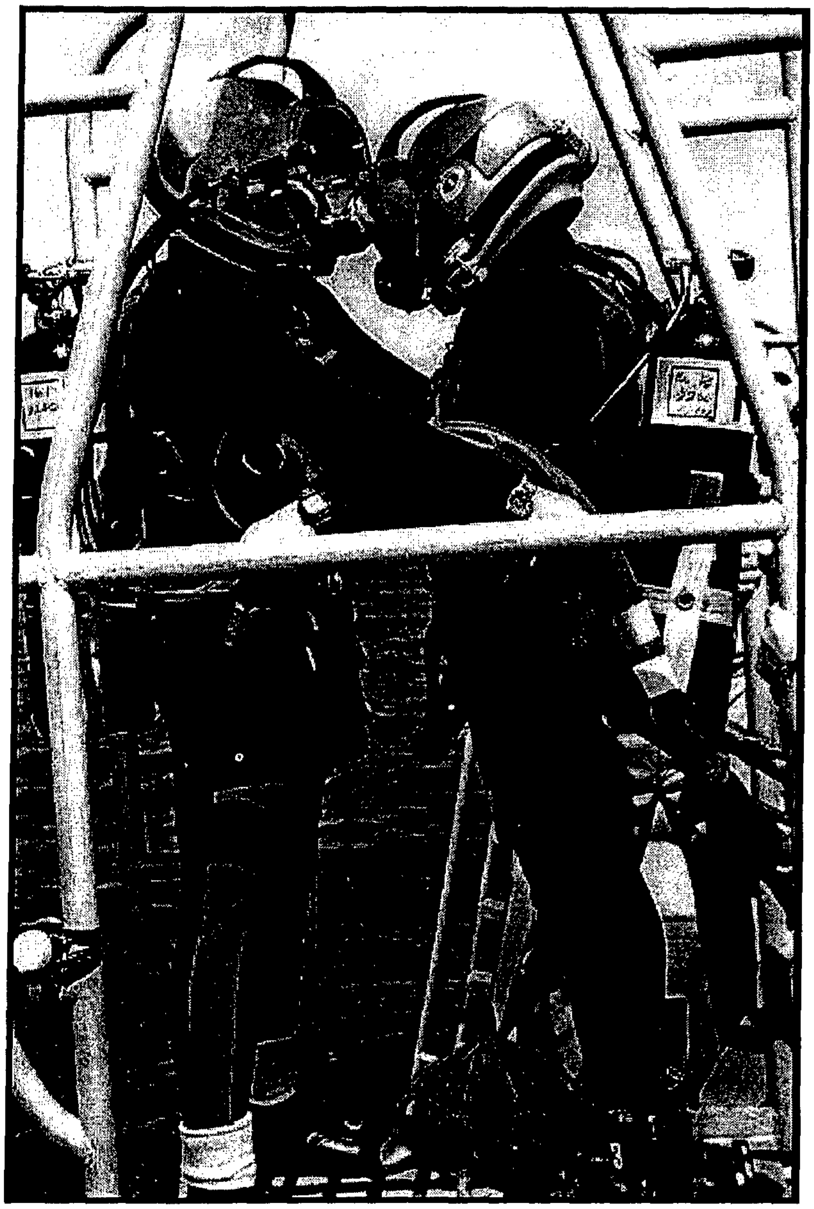

Figure 2. Two MK 21 divers on the stage, ready to descend.
(U.S. Navy Diving Manual, Rev. 4.)

I. MK 3 LWDS FLOW CALCULATIONS

METHODS

Approach

We use two modeling approaches to find the airflow rates that the MK 3 LWDS should support. We assume a worst case scenario, with all three divers on the bottom. Once we have confidence in those theoretical flow rates, we assess how those flow rates meet the divers' physiological requirements during both normal and emergency operations.

The first modeling approach is based on flow calculations for steady-state, isothermal flow of compressible gases in pipes. Various parameters derived from those calculations are then used to calibrate steady-state electrical models which then are expanded into non steady-state, dynamic electrical models. These dynamic models should give us the clearest picture to date of what a three-man dive team might experience during an emergency at the maximum certified depth for the MK 3 diving system.

Assumptions

To calculate pressure drops in the MK 3 LWDS console and diver umbilicals, we assume steady isothermal flow of ideal gases. The equations used for these calculations appear in Nuckols et al[2], and are applied in the *PipeFlow* program written by Nuckols himself. Many sources for pipe flow calculations are available, but *PipeFlow* is one of the more convenient for our purposes. We also assume that the nominal internal diameter of a 3/8 in. umbilical is 0.375 in.

Umbilical Pressure Losses

The manufacturer of diving hoses, Gates Rubber Co., has not tabulated hose friction (f) or roughness numbers. However, it provided the Table 1 information from which we are able to use *PipeFlow* to estimate internal hose roughness.

Table 1 shows the manufacturer-supplied data for the pressure drop across various lengths of a Gates 3/8-in. umbilical at a 250-psig inlet pressure. The temperature for these measurements is unknown, but we assume engineering standard at 68°F (20°C). (Another "engineering standard" temperature is 70°F. We use 68°F because of its historical use in LWDS calculations, shown on the following page.)

Table 1. Manufacturer-specified pressure drops for 3/8-in Gates umbilicals

Flow rate (scfm)	**Hose length (ft)**	**ΔP (psi)**
22	300	30.93
22	600	61.83
15	300	14.38
15	600	28.76

From these values, and using *PipeFlow's* standard engineering equations[2] we estimate an umbilical roughness (e) of $1.0 \cdot 10^{-4}$ ft. That degree of umbilical roughness allows us to predict pressure drop across the umbilical within an error of 6% or less. For comparison, standard tables of absolute roughness show a roughness of $1.5 \cdot 10^{-4}$ ft for wrought iron steel pipe, and $5.0 \cdot 10^{-6}$ ft for drawn tubing.

For 22 scfm flow at 68°F through a 3/8-in umbilical (3.3 acfm at 190 fsw), Reynolds number is 9.15×10^4, and the resulting $f = 0.028$. This friction factor is derived from the Colebrook formula for turbulent flow ($Re > 4000$).

$$\frac{1}{\sqrt{f}} = -2 \cdot \log_{10} \cdot \left[\frac{\frac{e}{D}}{3.7} + \frac{2.51}{\text{Re} \cdot \sqrt{f}} \right] \tag{1}$$

MK 3 Console Pressure Losses

With an estimate of umbilical roughness, we then described the flow characteristics of the MK 3 LWDS console as those resulting from flow through some unknown length of 3/8-in umbilical.

From an Analysis and Technology, Inc. (A&T), letter dated 12 August 1996 regarding LWDS Primary Air Circuit Flow Calculations, we find the following:

> At an inlet temperature of 68°F, and an inlet pressure of 265 psia on the console pressure gauge, and a flow rate of 65 scfm (enough to support 3 divers at approximately 22 scfm each), the pressure drop across Segment II of the console was 49 psi, such that outlet pressure (the upstream end of the umbilical) is 216 psia.

When we used that information in *PipeFlow*, we were able to describe the console internal plumbing as an equivalent length of 3/8-in I.D pipe with an assumed roughness of $1.0 \cdot 10^{-4}$ ft. That equivalent length was 58 feet.

The Operational Problem

The operational limits for the UBA MK 21 Mod 0 helmet are listed in the MK 21 Technical Manual[3] as 135 to 225 psi over ambient, with an optimum "overbottom" pressure of 185 psi. Tables 2 and 3 show the results of using *PipeFlow* to estimate the overbottom pressures available at a depth of 190 fsw at the outlet of 300-foot and 600-foot umbilicals with a minimal 15 scfm flowing to each of three divers. Total console flow is thus 45 scfm. Volumetric flow to each diver is 2.2 acfm since ambient pressure is 6.74 atmospheres absolute (atm abs) at 190 fsw. In Table 2 console pressure is 265 psia, and in Table 3 it is 235 psia.

Table 2. Approximate overbottom pressures at 190 fsw with 265 psia console pressure. and one of three 300-foot umbilicals at 190 fsw.

Length (ft)	scfm	e (ft)	Re	f	Pin (psia)	ΔP (psi)	Pexit (psia)	O/B (psi)
58 (console)	45	0.0001	187,000	0.027	265	22.5	242.5	---
300	15	0.0001	62,400	0.029	242.5	14.6	227.9	128
600	15	0.0001	62,400	0.029	242.5	30.2	212.3	113

In the first row of the table, we calculated the console exit pressure in psia. That then becomes the inlet pressure for the umbilicals. The pressure drop across the umbilicals are found in rows 2 and 3. For simplification, we ignore the effect of temperature differences along the length of the umbilical.

For a console pressure of 265 psia (Table 2) the calculated side block pressure ranged from 128 to 113 psi overbottom depending on umbilical length. For a console pressure of 235 psia (Table 3) side block pressure ranged from 93 to 74 psi, far below the minimal operating pressure of 135 psi overbottom.

Table 3. Approximate overbottom pressures at 190 fsw with 235 psia console pressure and one of three 300-foot umbilicals at 190 fsw.

Length (ft)	scfm	e (ft)	Re	f	Pin (psia)	ΔP (psi)	Pexit (psia)	O/B (psi)
58 (console)	45	0.0001	187,000	0.027	235	25.7	209.3	---
300	15	0.0001	62,400	0.029	209.3	17.1	192.2	93.1
600	15	0.0001	62,400	0.029	209.3	35.9	173.4	74.3

From these calculations, the only condition that is expected to provide a side block pressure close to the minimum operational pressure at 190 fsw is the use of 300-foot umbilicals with a 265 psia console pressure. This is of some concern since the MK 3 LWDS is authorized for use with console pressures as low as 235 psia during diving to 190 fsw (ref 4, pg. 8-2).

Due to the operational implications of these results, we thought it worthwhile to explore our calculation procedures more thoroughly, basically to see if we could obtain a more desirable result; i.e., one in better agreement with the Diving Manual. Specifically, we wanted to move towards a dynamic model that allows non-steady state gas flows.

65 scfm steady flow

A diver can supplement his air with the MK 21's steady flow valve. If a flow of 45 scfm through the MK 3 LWDS is marginal, as seen above, then a flow of 65 scfm should be even worse. Nevertheless, it is worth while to calculate the expected results for just such system flows because when steady flow valve are being used, the pressure requirements of the demand regulator do not apply. A flow of 65 scfm equals a flow at depth of 9.6 acfm, which will support three divers at a steady flow of 3.2 acfm each, or one at 5 acfm and two at 2.3 acfm.

If a diver's helmet floods, he may well want 5 or more acfm available when he operates the MK 21's steady flow valve. The question is, can he get 5 acfm when the pressure drops across long umbilicals are considered?

We already know from the A&T data above that with 65 scfm passing through the MK 3 console, the console outlet pressure is 216 psia, which also is the umbilical inlet pressure. We then compute the pressure drops across 300-ft and 600-ft umbilicals for a steady flow of 5 scfm.

To have 5 acfm at 190 fsw, 34 scfm must be flowing into the diver's umbilical. The following table shows the computed outlet pressures for 300- and 400-ft umbilicals, along with other parameters all obtained from *PipeFlow*.

Table 4. Umbilical outlet pressure predictions for a 5 acfm flow at 190 fsw.

Length (ft)	scfm	e (ft)	Re	f	Pin (psia)	ΔP (psi)	Pexit (psia)	O/B (psi)
300	34	0.0001	141,000	0.028	216	103.6	112.4	13.1
400	34	0.0001	141,000	0.028	216	179.9	36.1	-63.0

We conclude from Table 4 that 5 acfm at 190 fsw (34 scfm) cannot be delivered to the diver through a 400-ft umbilical, even with the steady flow valve wide open. It certainly could not be delivered through a 600-ft umbilical. We do not know exactly how much overbottom pressure is required to maintain flow through the side block assembly, steady flow valve, and gas train assembly in the MK 21 helmet, but an overbottom pressure of 13.1 psi at the end of a 300-ft umbilical may be marginal at best.

Table 5. Umbilical outlet pressure predictions for a 3.3 acfm flow at 190 fsw.

Length (ft)	scfm	e (ft)	Re	f	Pin (psia)	ΔP (psi)	Pexit (psia)	O/B (psi)
300	22	0.0001	91,500	0.028	216	36.6	179.4	80.3
600	22	0.0001	91,500	0.028	216	83.0	133.0	33.9

A standard flow rate of 22 scfm equates to a flow of 3.3 acfm at 190 fsw. As shown in Table 5, that flow rate should be maintainable in both 300- and 600-ft umbilicals, if we assume that an overbottom pressure of 34 psi is enough to drive gas through the gas supply train with the steady flow valve open.

In another case we can calculate the sequential pressure drops across the MK 3 console for an 80-scfm flow and a single 300-ft umbilical with 1/3 of 80, or 26.7 scfm, flowing (Table 6). At 190 fsw, the flow from each umbilical is 4.0 acfm (26.7 scfm/6.74 atm). As indicated by the low 17-psi overbottom pressure at the end of the umbilical, the flow rate of 4.0 acfm is about the maximum the system can sustain for steady flow operation.

Table 6. Approximate maximum flow through console and one of three 300-foot umbilicals at 190 fsw.

Length (ft)	scfm	e (ft)	Re	f	Pin (psia)	ΔP (psi)	Pexit (psia)	O/B (psi)
58 (console)	80	0.0001	333,000	0.027	265	79.3	185.7	---
300	26.7	0.0001	111,000	0.028	185.7	70.0	115.7	16.7

Even the low 17-psi overbottom pressure is not achievable in slightly longer umbilicals. For instance, in 400-ft umbilicals, the umbilical outlet pressure is 80 psia, less than the ambient pressure of 99 psia at 190 fsw.

As shown in Table 7, at 190 fsw the maximum flow sustainable in each of three 600-ft umbilical is estimated to be 3.5 acfm (23.3 scfm/6.74 atm). Since overbottom pressure is less than 3 psi, that flow rate is obtainable only in an open umbilical at 190 fsw, without a helmet.

Table 7. Estimated maximum flow through console and one of three 600-foot umbilicals at 190 fsw.

Length (ft)	scfm	e (ft)	Re	f	Pin (psia)	ΔP (psi)	Pexit (psia)	O/B (psi)
58 (console)	70.0	0.0001	291,000	0.027	265	58.1	206.9	---
600	23.3	0.0001	97,000	0.028	206.9	105.3	102.0	3.0

Electrical Models:

The foregoing presumes a steady state system. Arguably, the most relevant behavior of the MK3 LWDS system occurs in the unsteady state. We modeled unsteady system dynamics by taking the information obtained from the above steady state analysis and using it in a dynamic model analyzed by electrical circuit simulation. For this analysis we used MicroSim PSPICE (ver. 8.0) modeling system by MicroSim Corporation (Irvine, CA) (now owned by Cadence Design Systems, Inc., Irvine CA) running on a PC using Windows XP rather than the Windows 95 or Windows NT software for which it was designed.

PSPICE is able to model nonlinear resistors, in which resistance varies as a function of flow rate. And indeed, if we recalculate the data of Table 1 and add a flow resistance column, we see the following (Table 8):

Table 8. Calculated umbilical resistance for 3/8 in. Gates umbilicals.

Flow rate (scfm)	**Hose length (ft)**	**ΔP (psi)**	**Pamb (atm abs)**	**R (psi/scfm)**
22	300	30.93	1.0	1.41
22	600	61.83	1.0	2.81
15	300	14.38	1.0	0.96
15	600	28.76	1.0	1.92

For any given umbilical length, resistance at 22 scfm is almost 1.5 times greater than it is at a flow of 15 scfm. To better quantify this relationship, flow resistance (R) is expressed as an empirical function of umbilical length (Lumb), ambient pressure in atm abs (Pamb), and flow rate ($\dot{V}$).

$$R = a \cdot Lumb \cdot Pamb \cdot \dot{V}^n \qquad (2)$$

The data provided in Table 8 is then used to solve a system of simultaneous equations with two unknowns, "a" and "n." When Lumb is in feet and $\dot{V}$ is in scfm, a = $2.112 \cdot 10^{-4}$ and n = 1.004.

The exponent of flow (n) is very close to 1.0. In other words, R is linearly related to flow rate. However, as a didactic aid, our first electrical models assume that R is proportional to Lumb and Pamb only. The estimates of flow rate and pressure drops can then be corrected with flow-dependent impedances before we move to the dynamic models.

Choice of Resistance Values

As Appendix C shows, when we model a pneumatic system by electrical analogs, we must carefully choose the resistance values used in the model. We use resistance values based on flow in standard units.

We know that the pressure drop across the MK 3 console is 49 psi when 65 scfm of air is flowing through it. Therefore, our best estimate for console resistance is 49 psi/65 scfm or 0.75 psi/scfm. We therefore use 0.75 ohms (Ω) as a first approximation of console flow resistance.

From Table 5 we see that for a 22-scfm flow (3.3-acfm flow at 190 fsw) through a 600-ft umbilical, the pressure drop is 83 psi. Accordingly, umbilical flow resistance (Rumb) at 22 scfm is 3.77 psi/scfm (Ω). This resistance value is higher than the 2.81 psi/scfm shown in Table 8, for reasons explained in Appendix C. Likewise, for 300-ft umbilicals, ΔP = 36.6 psi. For 22 scfm, R = 1.66 Ω, which again differs predictably from the 1.41 Ω given in Table 8.

The Model — Constant Resistance

Figure 3 presents a steady-state electrical model of the entire MK 3 system with the previously found component resistance values used and three 600-ft umbilicals attached. (Throughout this report we use the convention of umbilical number 1 being in the middle, number 2 below, and number 3 above umbilical number 1.) Source voltage (Esource) has the same magnitude as absolute pressure in psi, and current in amperage (A) has the same magnitude as flow rate in cubic feet per minute. In this case, the pressure at the end of each umbilical is 133 psi, 34 psi above ambient pressure at 190 fsw. These analytical results from the electrical model of Figure 3 closely match the flow calculations for 600-ft umbilicals in Table 5.

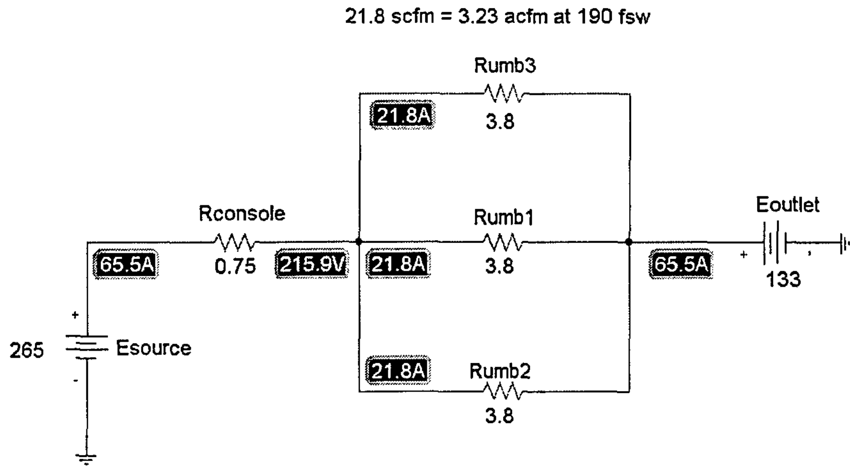

Figure 3. Three 600-foot umbilicals in parallel with umbilical outlet pressure = 133 psi. At 190 fsw umbilical flow is equivalent to 3.2 acfm.

We can estimate the maximum flow achievable at 190 fsw by allowing outlet pressure to equal ambient pressure at 190 fsw (Figure 4). That maximum flow is about 4.1 acfm. Of course, that maximum flow is not particularly useful, since overbottom pressure at the end of the umbilical is zero psi: no pressure is available to force gas through the MK 21 gas train. This maximal flow is also an underestimate, since flow resistance actually increases as flow increases (Equation 2, Table 8), a fact unacknowledged in this example. Nevertheless, this calculation helps bound the range of flows available in 600-ft umbilicals at 190 fsw.

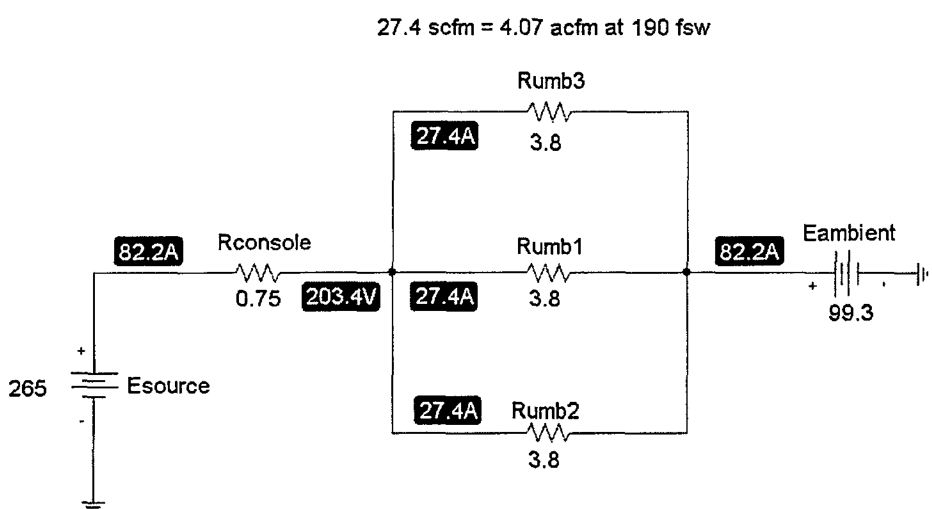

Figure 4. Maximal flows in three 600-foot umbilicals in parallel at 190-fsw. Overbottom pressure is zero.

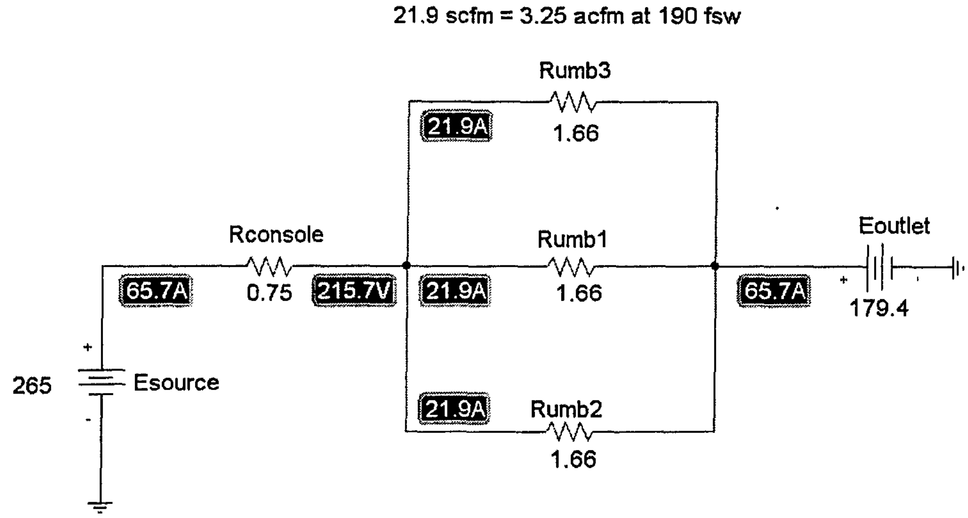

Figure 5. Three 300-foot umbilicals in parallel.

With about 66 scfm flowing through the console and three 300-ft umbilicals, umbilical outlet pressure should be about 81 psi overbottom pressure (Figure 5). Not unexpectedly, this result corresponds to the flow calculations for 300-ft umbilicals in Table 5.

The maximal flow rate with zero overbottom pressure is roughly 6.3 acfm (Figure 6), which represents the absolute upper bound for flow rate. Again, that flow is too high an estimate, since no flow-dependent resistance is considered. Furthermore, a zero psi overbottom pressure value is not useful, except to estimate the gas flowing from a severed umbilical.

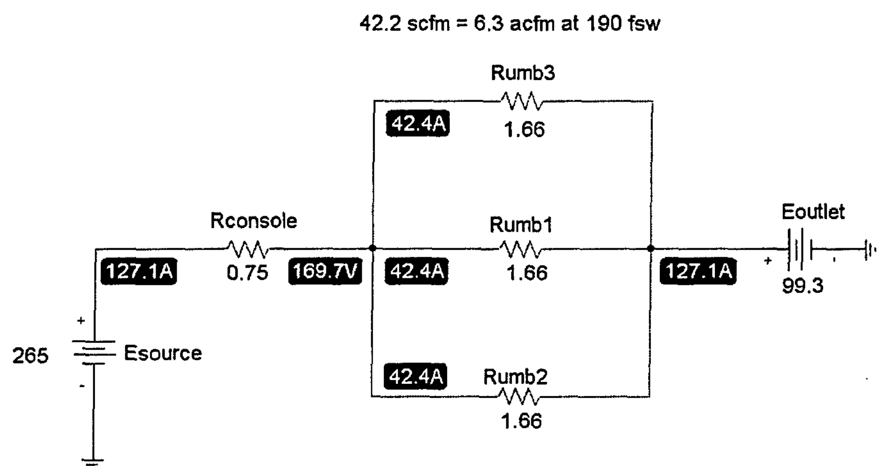

Figure 6. Maximal flows in three 300-foot umbilicals in parallel. The same caveats apply as in Figure 4.

The Model — Flow-Dependent Impedance

Figure 7 shows the schematic of Figure 4 but with the umbilical resistances (Rumb) converted into flow-dependent impedances (ZX). In PSPICE the value of each ZX is determined by the product of two values: that of a fixed-reference resistor (Rref), which represents umbilical length and other flow-independent factors, and that of a voltage-dependent part. In this case, the voltage is that of a drop across a 0.01 Ω resistor and thus varies directly with flow rate. The greater the flow through the 0.01 Ω resistor in series with each umbilical and helmet, the greater the voltage drop, and the greater the flow-dependent portion of umbilical impedance.

The ZXs are so adjusted that the data from Table 8 can be reproduced for an E source of 250 psig. For a condition in which 22 scfm flows through each 600-ft umbilical, the pressure drop across each umbilical's 0.01 Ω resistor is 0.22 V. For a flow of 15 scfm (A), the voltage drop is 0.15 V. Multiplying the respective pressure drops by the reference resistor value of 12.73 Ω yields a ZX of 2.80 Ω, a total impedance of 2.81 Ω for the 22 scfm flow in a 600-ft umbilical. When flow is 15 scfm, total impedance is 1.92 Ω, just as shown in Table 8.

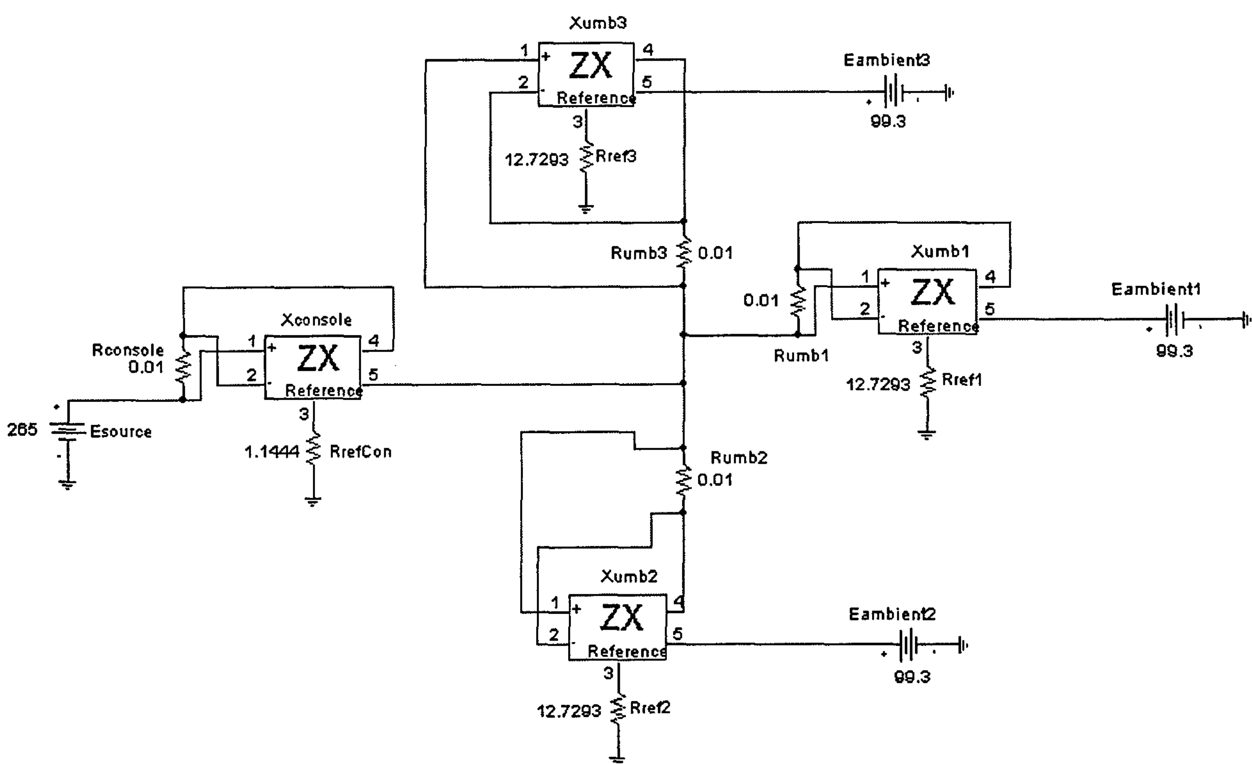

Figure 7. Flow dependent impedances (ZX) in 600-foot parallel umbilicals.

When a Rref of 6.345 Ω is used to model 300-ft umbilicals, umbilical impedances of 1.41 Ω and 0.96 Ω result for the 22 scfm and 15 scfm flows, respectively. In a similar manner, the Rref for the MK 3 console is 1.14 Ω.

Once the model is calibrated in this manner, we can rely upon it to produce appropriate impedances and pressure drops over a range of umbilical flow rates. Table 9 shows the results of exercising the model of Figure 7 for 600-ft umbilicals, or a comparable model for 300-ft umbilicals. We simulated various console and downstream pressures.

Table 9. Flow dependent vs non-flow dependent impedance (steady state).

Length (ft)	Esource (psi)	Pamb (psi)	I_{ZX} (scfm)	I_{ZX} (acfm)	I (acfm)	ΔI (I_{ZX} - I)
600	265	99.3	26.7	4.0	4.1	-0.1
600	265	130	24.1	3.6	3.3	0.3
600	235	99.3	24.2	3.6	3.3	0.3
600	235	130	21.3	3.2	2.6	0.6
300	265	99.3	31.4	4.7	6.3	-1.6
300	265	116	29.8	4.4	5.7	-1.3
300	265	130	28.4	4.2	5.1	-0.9
300	235	99.3	28.4	4.2	5.2	-1.0
300	235	130	25.0	3.7	4.0	-0.3

Effect of flow-dependent impedance

Compared to the models with nonflow-dependent resistors, the models with flow-dependent impedances produced moderated flows. Flows across the conditions shown in Table 9 varied by 240% in the nonflow-dependent model (I), but varied by only 147% in the flow-dependent impedance models (I_{ZX}). Flow-dependent impedances tended to decrease flow in the already high 300-ft umbilicals, but mostly increased flow in the 600-ft umbilicals.

We also see from Table 9 that under the conditions examined, flows do not exceed 4 acfm in 600-ft umbilicals. Even in 300-ft umbilicals, when flow dependence of resistance is acknowledged, predicted flows (I_{ZX}) do not reach 5 acfm.

Dynamic Model with Two MK 21 Helmets in Demand Mode

The real power of electrical circuit simulation comes when modeling dynamic behavior of complex systems. For the MK 21 helmet, Figure 8 shows a relatively simple electrical model having no flow-dependent impedances in it. For the entire MK 3 LWDS with flow-dependent umbilical impedance, Figure 9 shows a more complex schematic.

In Figure 8 the MK 21 diver is represented by a sinusoidal voltage source (circle with a backward S, left-center of the figure). The portion of the circuit above the diver is the inhalation portion of the circuit; that below the diver represents the exhalation path. A negative mouth pressure generated by the diver causes a regulator to open and allow gas flow from the umbilical to pass through an inhalation resistance and a one-way valve modeled by a diode (Dinh) en route to the diver's mouth. A small positive mouth pressure causes another set of valves to open and dump gas through an exhalation resistance into the seawater.

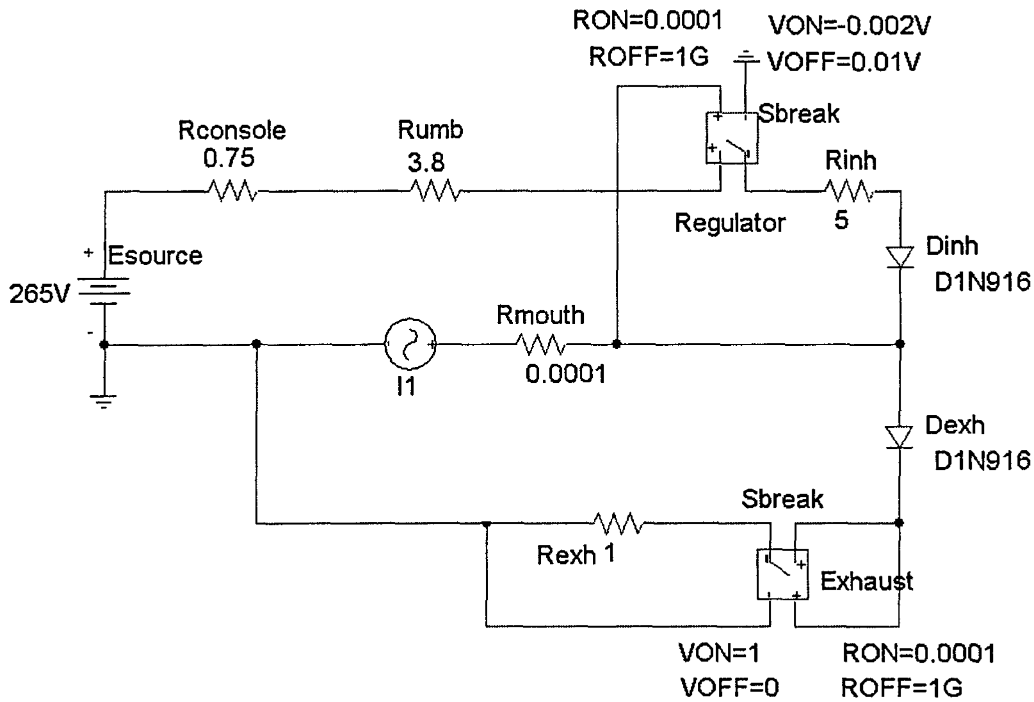

Figure 8. PSPICE schematic of a single MK 21 system..

Figure 9 shows the complete dynamic model in schematic form. The circuit shows helmets 2 and 3 on demand flow (helmet 3 on top), and helmet 1 (in the middle) with its steady flow valve wide open.

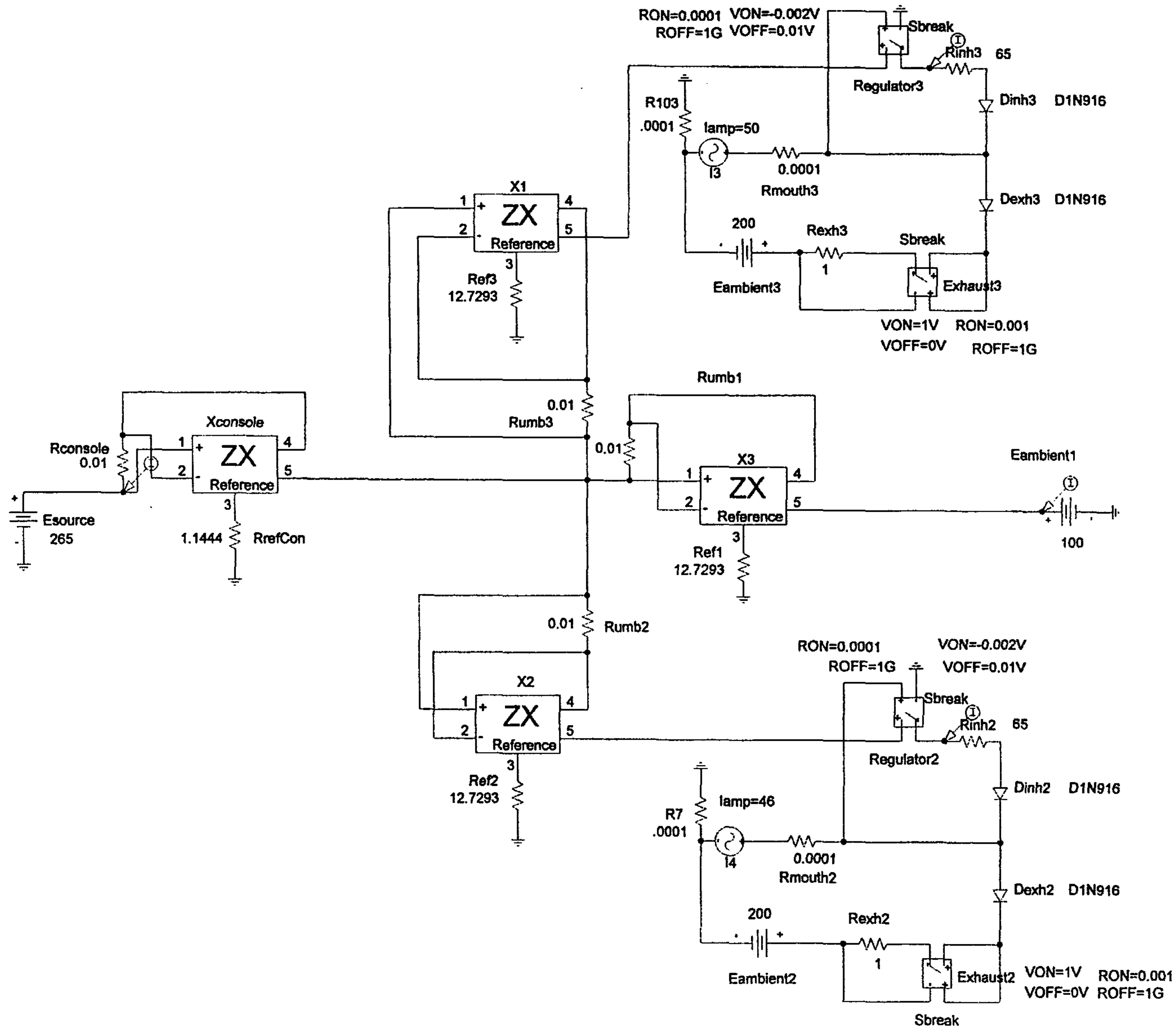

Figure 9. System schematic of a three-diver system for dynamic modeling, with 600-ft umbilicals and diver 1 on steady flow.

Figure 10 shows simulation results from the dynamic model for 600-ft umbilicals with umbilical outlet pressures of 200 psi for divers 2 and 3 on demand flow, and 100 psi for diver 1 on steady flow (represented by the solid line). The dashed and dotted lines represent inspiratory flows for divers 2 and 3.

The almost simultaneous breathing of divers 2 and 3 transiently but markedly reduced the steady flow available to diver 1. His so-called "steady flow" varied from 34.4 scfm to 14.3 scfm. The average flow was 32.9 scfm, or 4.9 acfm. The peak inspiratory flow rates for divers 2 and 3 averaged 48 scfm, or 7.1 acfm (3.35 L/s) at 190 fsw. Based on equation 5 and represented in Figure 11, that peak flow rate corresponds to an RMV of 64 L/min:

$$RMV = \frac{\dot{V}peak \cdot \frac{L}{s} \cdot 60 \cdot \frac{s}{\min}}{\pi}. \qquad (5)$$

When the breathing of divers 2 and 3 are not in phase, the periodic declines in steady flow are not as pronounced, but the average steady flow rate remaines unchanged.

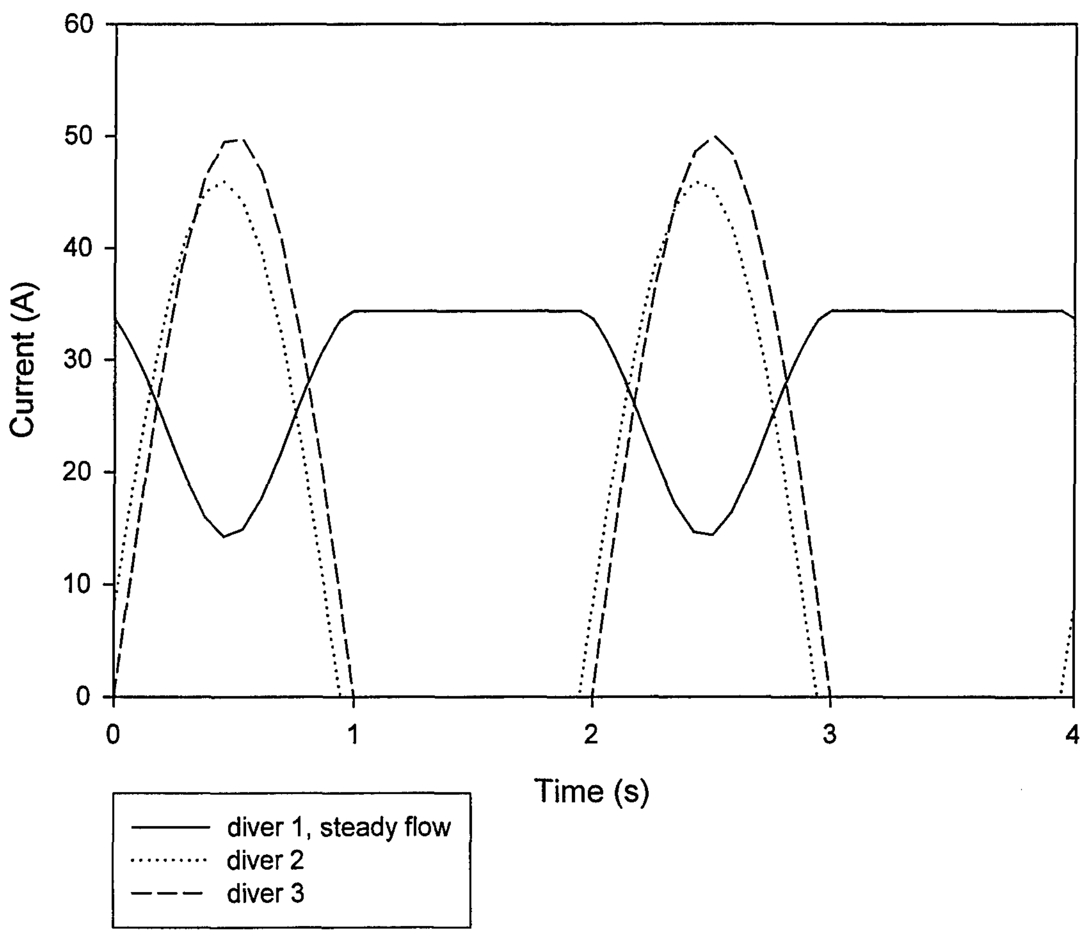

Figure 10. PSPICE simulation results for Figure 9.

When the average RMV for divers 2 and 3 was reduced to 40 L/min, the average steady flow for diver 1 increased only slightly from 4.9 to 5.0 acfm. The maximum steady flow was 5.1 acfm when divers 2 and 3 were not inhaling.

Equation 5 and Figure 11 show that an average steady flow rate of 5 acfm should enable the diver on steady flow to maintain an RMV of as much as 45 L/min without encountering gas insufficiency. Higher RMVs would require the diver to draw additional gas from other sources such as the helmet and neck dam region, or to alter his respiratory wave form — for instance, from sinusoidal to rectangular breathing.

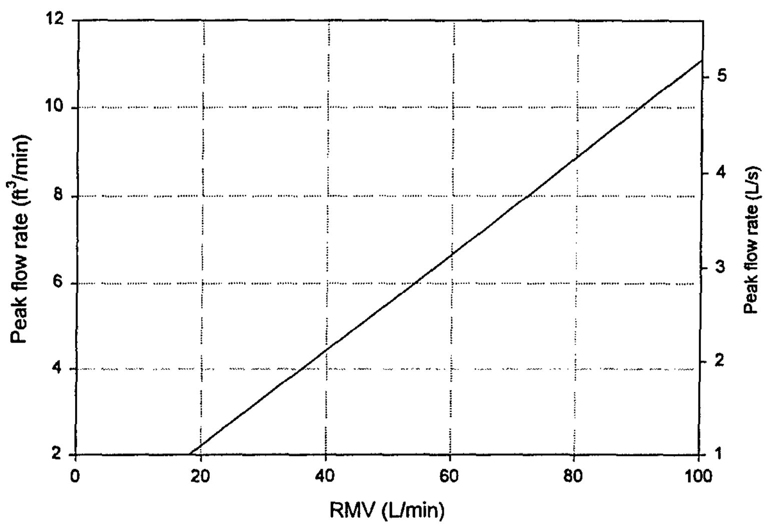

Figure 11. Relation between peak flow and average minute ventilation based on Equation 5.

Unfortunately, so-called "steady" flow may fluctuate considerably, as shown in Figure 10, depending on the breathing patterns of the two divers on demand flow. For diver 1, his ill-timed inspiration could cause respiratory distress and temporarily exacerbate helmet flooding.

Table 10 shows predictions for average steady flow based on the dynamic electrical model (Figure 9) with 600- and 300-ft umbilicals, as well as 600-ft umbilicals with a reduced driving pressure (235 psia). To put the resulting steady flow rates into perspective, the table also shows the RMV, oxygen consumption, and analogous swimming speed that the steady flow rates support. The estimates of oxygen consumption and swimming speed are from the U.S. Navy Diving Manual[3], pg. 3-12.

The average steady flow rates for umbilical 1 shown in Table 10 exceed the predicted flow rates in Table 9, where all umbilicals have steady flow. Presumably, these Table 10 rates are higher than Table 9's because umbilicals 2 and 3 in the dynamic model are not always flowing gas. During those pauses, console flow is diverted to umbilical 1, which needs a continuous, if not truly steady, flow.

Table 10. Steady flow (SF) results for the dynamic model (Figure 9).

Umb. Length (ft)	Esource (psia)	Poutlet (psia)	average SF (acfm)	RMV (L/min)	RMV (acfm)	$\dot{V}O_2$ (L/min)	Swim Speed (kts)
600	265	100	4.88	44.0	1.6	1.7	1
300	265	100	6.63	59.8	2.1	2.5	1.2
600	235	100	4.43	39.9	1.4	1.4	0.8

Umb. = umbilical, Esource = driving voltage (pressure), Poutlet = outlet pressure at the end of the umbilical, SF = "steady" flow. $\dot{V}O_2$ and swim speed from the U.S. Navy Diving Manual (pg. 3-12.)

Comparison to NEDU TR 12-95

The above models did not include the flow resistance for the MK 21 helmet on steady flow. Consequently, the values for steady flow estimated from the model are presumably higher than those measured in actual MK 3 systems. Indeed, the average steady flow values in Table 10 are higher than the maximum steady flow rates reported in reference 1 (Fig.12). Our estimated values for 600-ft umbilicals are only slightly higher than values given in the 1995 report. However, our estimates for flow in 300-ft umbilicals are considerably higher than those previously reported.

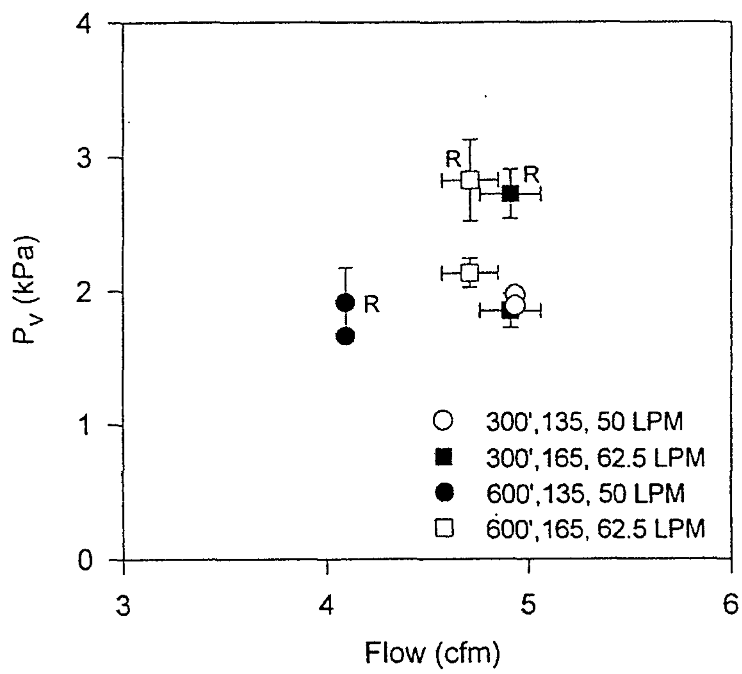

Figure 12. Maximum steady flows reported in reference 1. Figure captions list umbilical length, console pressure, and RMV in L/min for both helmets in demand mode. Console pressures are reported as gauge pressure above ambient at 190 fsw.

Maximum Steady Flow Rate Summary

Table 11 summarizes the maximum flow predictions for the MK 3 LWDS console combined with either 300- or 600-ft umbilicals. These models assume that the end of the umbilical is at 190 fsw and that either the MK 21 helmet is not present, or its internal flow resistance is negligibly small.

Table 11. Maximum Steady-Flow Rate Summary.

Source	300-foot umbilical	600-foot umbilical
PipeFlow	4 acfm (Table 6)	3.5 acfm (Table 7)
Steady Flow electrical (no ZX) (Table 9)	6.3 acfm	4.1 acfm
Steady Flow electrical (ZX) (Table 9)	4.7 acfm	4.0 acfm
Dynamic Model (Table 10)	6.6 acfm	4.9 acfm
NEDU 12-95, Fig. 12	4.9 acfm	4.7 acfm

PipeFlow, the steady flow electrical model with ZX components, and the Dynamic Model all contained non-linear resistances. The first two models, being steady state models. matched the steady state, nonlinear umbilical flow data shown in Tables 1 and 8. The predictions of those models differ by no more than 15%, with the electrical model predicting higher flows than the mechanical modeling of *PipeFlow.*

The flows estimated from the dynamic model are higher than in the other models, presumably because airflow to divers 2 and 3 is intermittent. It matches the experimental data shown in Figure 12 for 600-foot umbilicals, but considerably overestimates the flow available with 300-foot umbilicals.

DISCUSSION

EGS Use

Much of the above analysis and discussion becomes moot if the diver suffers a gas supply failure and simply switches to the EGS (emergency gas supply) while making an ascent. However, for safety's sake we have assumed a more problematic scenario with a MK 21 helmet failure that causes a large amount of water to flood into the helmet. Such an incident is unlikely to be resolved by using the EGS, simply because of the difficulty in getting a watertight seal between the diver's face and the helmet's oro-nasal mask. Such a case allows no alternative to using the steady flow valve to attempt helmet dewatering.

Comparison between Fluid Dynamic and Electrical Models

Electrical circuit simulation has been the preferred method for solving the systems of differential equations describing respiratory mechanics ever since the days of analog computers. The results of using electrical analogs to model breathing have proven to be robust and accurate. It is thus no surprise that in this analysis the steady state fluid dynamic and steady state electrical simulations are in many respects comparable.

Furthermore, the equivalence between voltage and pressure, between gas flow and current, between electrical and flow resistance is so clear mathematically that they can be used interchangeably, as they have been here. (Appendix C illustrates some cases where interchangeability may lead to modeling difficulties.)

The most complete steady state models — the mechanical *PipeFlow* model and the electrical steady-state, nonlinear resistance models — predicted similar maximum flow rates at 190 fsw. Not surprisingly, the dynamic models predicted higher maximum flow rates for the diver on steady flow, since the divers on demand flow were inhaling only half the time and thus putting a load on the gas supply only half the time.

The major advantage of a dynamic model over a simple steady-state model is that dynamic models allow us to visualize gas stealing among the divers. We see that stealing can cause large but transient drops in continuous flow to the diver with an open steady flow valve under the conditions analyzed (Fig. 10). However, because the stealing is transient, the average continuous flow is not seriously impaired.

Results from NEDU 12-95

The possibility has been raised that NEDU TR 12-95 may have overestimated the maximum steady flow rates available from the MK 3 LWDS at 190 fsw. That concern was one of the rationales for the detailed analyses reported here. The results from the *PipeFlow* software (Table 11) seemed to reinforce that concern, since maximum steady flows were calculated to be considerably less than the reported measurements[1]. The steady-state electrical models which followed the *PipeFlow* analyses again emphasized that concern, at least for 600-ft umbilicals. It was not until the dynamic electrical model was generated did theory seem to match the 1995 flow measurements, at least for 600-ft umbilicals. Unfortunately, the dynamic model raised the question of why measured flows in 300-ft umbilicals were so low, relative to measured flows in 600-ft umbilicals.

Orifice Equations

Explanations why the agreement between measured and modeled flows in 300-ft umbilicals is so much poorer than in 600-ft umbilicals are few. One potential explanation is that total flow to the diver is higher with 300- than with 600-f umbilicals, and the flow dependence of impedance may be greater than estimated. In other words, impedance (flow resistance) through MK 21 helmets might be related to flow to the second power instead of to the first power, as it is for the umbilicals (Table 8, Equation 2). We examine that possibility below.

The most likely resistance-producing flow regime in a diving helmet is orifice flow, which is described by the equation

$$q = a \cdot Y \cdot d^2 \cdot \frac{C}{Sg} \cdot \sqrt{\Delta P \cdot \rho} \qquad (6)$$

where q = flow rate at standard conditions, Y = net expansion factor for compressible flow through orifices, d = orifice diameter, C = flow coefficient (discharge coefficient corrected for convective acceleration), Sg = specific gravity relative to air, ΔP = pressure drop across the orifice, and ρ = gas density.

This equation can be rearranged to find the flow dependence of resistance for orifice flow.

$$R = \frac{\Delta P}{q} = q \cdot \frac{Sg}{\rho \cdot \left(a \cdot Y \cdot C \cdot d^2\right)^2} \qquad (7)$$

The flow coefficient (C) for square edge orifices is somewhat dependent upon Reynolds number, but C can both increase and decrease as Reynolds number increases. Since the other factors in the orifice flow equation are not flow dependent, we conclude that, to a first approximation, orifice flow resistance is linearly proportional to flow rate (q), just as in the case of umbilical flow.

At this time we cannot explain why resistance estimates for 300-ft umbilicals considerably exceed those resistances measured.

Summary

Certification requirements for the MK 3 LWDS require its console to provide a flow of 3.2 acfm when a back pressure equal to that of a set of three 600-ft umbilicals is connected to the console. According to Table 5, the *PipeFlow* model of this steady flow test suggests that 3.3 acfm can be easily achieved at an exit pressure equal to that of 190 fsw in both 300- and 600-ft umbilicals.

From Table 9 we see that the steady flow electrical models with variable ZX also predict that 3.3 acfm can be easily achieved at 190 fsw, even with a console pressure as low as 235 psia.

For real-world diving, however, a more important requirement is that a stricken diver on steady flow be able to reach a high flow rate even with two other divers on the bottom in demand mode. The steady flow models are of little benefit in investigating that scenario. From Table 10 we see that with 600-ft umbilicals, a diver at 190 fsw, and console pressure at 265 psia, steady flow available to the stricken diver should be approximately 4.9 acfm, in reasonable agreement with data published in NEDU TR 12-95 (~4.7 acfm). This gas flow should permit an RMV of 44 L/min.

The dynamic model predicts that heavy ventilatory rates (50-90 L/min) could not be sustained by a stricken diver breathing off of his steady flow at the end of a 600-foot umbilical at 190 fsw. These RMVs could not be sustained with a 265 psia console pressure, and certainly not with a 235 psia console pressure. High ventilatory rates in response to heavy work or panic would expose the diver to some or all of the following: high respiratory pressures, severe CO_2 retention, or accelerated helmet flooding.

Based on data reported in NEDU TR 12-95, dynamic model predictions at 190 fsw with 300-ft umbilicals may be overoptimistic. The discrepancy between model predictions for 300-ft umbilicals and the data presented in the 1995 report are so far unexplained.

Recommendations

If a MK 3 console can provide only 135 psia over bottom pressure at 190 fsw, it should only be used with considerable caution with 600-ft umbilicals. The system's ability to supply enough steady flow to provide breathing gas for two divers at 190 fsw and dewater the helmet of a third diver at that depth would be limited.

If a diver at the end of a 600-ft umbilical is attempting to breathe off his steady flow at 190 fsw, even with the console pressure at 165 psig overbottom pressure, he should minimize his workload and allow his buddy and standby to assist him. Strenuous work on his part could cause gas starvation, resulting in panic, unconsciousness, or helmet flooding.

II. Flow Measurement Devices

In selecting a flow meter that would be best suited to measure diving console output, we reviewed the following classes of flow meters:

- coriolis
- differential pressure
  - laminar flow elements
  - orifice
  - pitot-static tube
  - venturi
- positive displacement
- turbine
- vortex shedding
- area meters (Rotameter)
- V-element meter
- target meter
- ultrasonic
- thermal

Descriptions and characteristics of many of these flowmeter types are given in Appendix B.

Our selection criteria for the best flowmeter for use in measuring gas flows from gas supply consoles, such as the MK-3, included ability to handle compressed gases, ease of use, robustness, sensitivity within the expected Reynolds number range, relative insensitivity to contamination, plus minimal maintenance, calibration requirements and cost.

On the basis of all of the above criteria, the best flow measurement device would appear to be an orifice meter. Details of the use and calculations involving orifice flow meters are given below.

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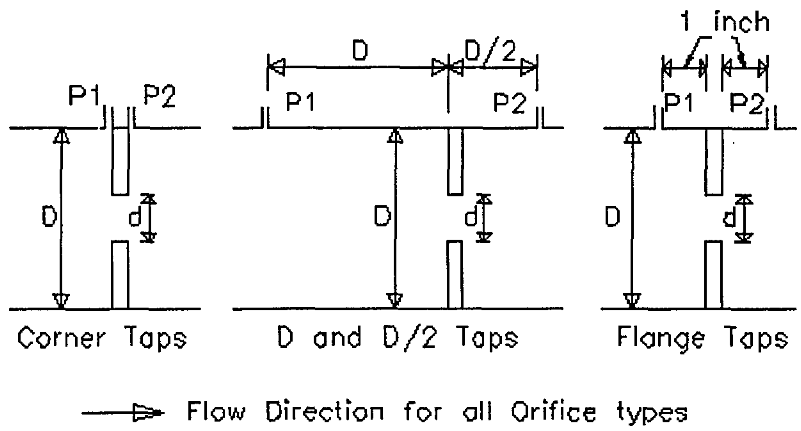
Types of Pressure Taps for Orifices:



Orifice flowmeters are used to determine a liquid or gas flow rate by measuring the differential pressure (P1 - P2) across the orifice plate. Orifice meters are typically used in 5 cm to 1 m diameter pipes. They are generally less expensive to install and manufacture than the other commonly used differential pressure flowmeters; however, nozzle and venturi flow meters have the advantage of lower pressure drops. Equations for orifice meters have the advantage of no Reynolds Number upper limit for validity.

An orifice flowmeter is typically installed between flanges connecting two pipe sections (flanges are not shown in the above drawings). The three standard pressure tapping arrangements are shown in the drawings; the location of the pressure taps affects the discharge coefficient somewhat. Flange pressure taps penetrate the flange and are at a standard distance of 1 inch (2.54 cm) from either side of the orifice. For corner taps or D and D/2 taps, the pressure tap locations are as shown.

Orifices are typically less than 0.05D thick. For exact geometry and specifications for orifices, see ISO (1991)[5] or ASME (1971)[6]. The ASME and ISO have been working on guidelines for orifices since the early 1900s. The organizations have the most confidence in orifice accuracy when the Reynolds number exceeds $1 \cdot 10^5$, though Reynolds numbers as low as $4 \cdot 10^3$ are valid for certain d/D ratios as discussed below. The calculation above is for flow of gases. For liquid flow through orifices, please visit our orifice calculation for liquids. Gas flow calculations include an expansibility factor (e), which is not present in the liquid calculation. The expansibility factor accounts for the effect of pressure change on gas density as gas flows through the orifice.

Equations

The following calculations are for orifices carrying a gas as described in ISO (1991 and 1998)[5,7].

$$Q_m = \frac{eCA_{throat}\sqrt{2\rho\Delta p}}{\sqrt{1-\beta^4}} \qquad Q_a = \frac{Q_m}{\rho} \qquad Q_s = Q_a \frac{P_1 T_{std}}{P_{std} T}$$

$$\rho = \frac{P_1}{RT} \qquad e = 1-(0.41+0.35\beta^4)\frac{\Delta P}{KP_1} \qquad \beta = \frac{d}{D}$$

$$\mathrm{Re}_D = \frac{V_{pipe} D}{\nu} \qquad \mathrm{Re}_d = \frac{V_{throat} d}{\nu} \qquad \nu = \frac{\mu}{\rho}$$

$$w = \frac{\sqrt{1-\beta^4} - C\beta^2}{\sqrt{1-\beta^4} + C\beta^2}\Delta P \qquad K_m = \frac{2w}{\rho V_{pipe}^2} \qquad V_{pipe} = \frac{Q_a}{A_{pipe}}$$

$$V_{throat} = \frac{Q_a}{A_{throat}} \qquad A_{pipe} = \frac{\pi}{4}D^2 \qquad A_{throat} = \frac{\pi}{4}d^2$$

Discharge Coefficient:

Corner Pressure Taps: L1 = L'2 = 0

$$C = 0.5961 + 0.0261\beta^2 - 0.216\beta^8 + 0.000521\left(\frac{10^6\beta}{\mathrm{Re}_D}\right)^{0.7}$$

$$+\left(0.0188 + 0.0063\left(\frac{19000\beta}{\mathrm{Re}_D}\right)^{0.8}\right)\left(\frac{10^6}{\mathrm{Re}_D}\right)^{0.3}\beta^{3.5}$$

$$+\left(0.043 + 0.08e^{-10L_1} - 0.123e^{-7L_1}\right)\left(1 - 0.11\left(\frac{19000\beta}{\mathrm{Re}_D}\right)^{0.8}\right)\frac{\beta^4}{1-\beta^4}$$

$$-0.031\left(\frac{2L'_2}{1-\beta} - 0.8\left(\frac{2L'_2}{1-\beta}\right)^{1.1}\right)\beta^{1.3}$$

If $D < 0.07112$ m (2.8 inch), then add the following (where D is in meters):

$$+0.011(0.75-\beta)\left(2.8 - \frac{D}{0.0254}\right)$$

D and D/2 Pressure Taps: L1 = 1 and L'2 = 0.47
Flange Pressure Taps: L1 = L'2 = 0.0254/D where D is in meters

Variables:
Dimensions: F = Force, L = Length, M = Mass, T = Time, t = temperature

Apipe =Pipe Area [L2], *Athroat* =Throat Area [L2], *C*=Discharge Coefficient
d=Throat Diameter [L], *D*=Pipe Diameter [L], *e*=Gas Expansibility
k=Equivalent Roughness of Pipe Material [L]
K=Gas Isentropic Exponent, *Km*=Minor Loss Coefficient
M=Mass Flowrate [M/T]
P_1 =Upstream Absolute Pressure [F/L2], P_2 =Downstream Absolute Pressure [F/L2]
ΔP =Differential Pressure [F/L2] = P_1-P_2
Pstd =Standard Absolute Pressure =14.73 psia = 1.016×10^5 N/m2
Qa =Actual Volumetric Flowrate [L3/T]
Qs =Volumetric Flowrate at Standard Pressure and Temperature [L3/T]
R=Gas Constant (used to compute gas density) = 8312/*W* N-m/kg-K
Re_d =Reynolds Number based on d, Re_D =Reynolds Number based on D
T=Gas Temperature [t] (converted automatically to absolute)
Tstd =Standard Absolute Temperature=520 °R=288.9K
Vpipe=Gas Velocity in Pipe [L/T], *Vthroat* =Gas Velocity in Throat [L/T]
w=Static Pressure Loss [F/L2], *W*=Molecular Weight of Gas [gram/mole]
β =Ratio *d/D*, ρ =Gas Density [M/L3], μ =Gas Dynamic Viscosity [F-T/L2]
ν =Gas Kinematic Viscosity [L2/T]

Validity and Discussion:

For all types of pressure taps: d >= 1.25 cm, 5 cm <= D <= 1 m, 0.1 <= d/D <= 0.75

For Corner Pressure Taps or D and D/2 Pressure Taps:
ReD >= 4000 for 0.1 <= d/D <= 0.5 and ReD >= 16,000(d/D)2 for d/D>0.5

For Flange Pressure Taps: ReD >= 4000 and ReD >= 170,000 D (d/D)2 where D is in meters

The calculation does not provide results if these values are out of range.

<u>Expansibility:</u>
The equation shown above for expansibility, e, is valid for P2/P1 >= 0.75. Our calculation gives a warning message if P2/P1 < 0.75, but still computes answers.

ISO Pipe Roughness Recommendation:

ISO recommends that in general $k/D <= 3.8x10^{-4}$ for Corner Taps and $k/D <= 10^{-3}$ for Flange or D and D/2 pressure taps. k is the pipe roughness.

Pressure Loss:

w is the static pressure loss occurring from a distance of approximately D upstream of the orifice to a distance of approximately 6D downstream of the orifice. It is not the same as differential pressure. Differential pressure is measured at the exact locations specified in ISO (1991)[8] (shown in the above figures).

Minor Loss Coefficient:

Km is computed to allow you to design pipe systems with orifices and incorporate their head loss. Head loss is computed as h=KmV2pipe/2g.

Standard Volumetric Flowrate:

Standard volumetric flow rate, Qs, is the volumetric flow rate computed at standard pressure and temperature, Pstd and Tstd (shown above in variables). Actual flow rate, Qa, is computed at the gas's actual pressure and temperature. Qs is useful to users who need to compute (or input) standard flow rate; often pump curves and flow measurement devices provide standard, rather than actual, flow rate. The advantage of using standard flow instead of actual flow is that the same device (or pump curve) can be used for a gas at various temperatures and pressures without re-calibrating for an infinite range of actual pressures and temperatures. The user can easily convert standard to actual flow rate if the actual temperature and pressure of the gas are known; our calculation does this automatically.

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APPENDIX A. Summary of MK 3 LWDS Flow Related NEDU Publications

NEDU TM90-10. Evaluation of the MK 20 and MK 21 UBA Supported by the MK 3 LWDS. K Hodina.

NEDU TM 91-10, Evaluation of the Gates 33 HB 3/8 in ID Divers Umbilical for Use with the MK 20 and MK 21 Mod 1 UBA. M Ross.

NEDU Report 12-95. MK 3 LWDS Air Flow Capacity. D Baiss, D Southerland, J Clarke. In response to NAVSEA Task 94-028. Deeper than 130 fsw, console O/B pressure should be increased to 165 psig, regardless of whether 300 or 600 ft long, 3/8 in. umbilical is used. Will support 2 divers with an RMV of 50 L/min (moderately hard work), and a third in intermittent steady flow.
Noted benefit of 600 ft umbilical gas storage volume on resistive effort.

11 Dec 96: NAVSEA Task 96-39: Diver ventilatory requirements, gas supply performance specification and standardized test requirements for divers life support systems (M Knafelc/R Mazzone).

1) Define diver ventilatory requirements
2) Define gas supply performance specification
3) Define and verify standardized test procedures to measure DLSS systems ability to meet performance specification.

5 Jun 97: MK 3 LWDS, MK 21 Helmet Resistive Effort (Pva) Results
Answer to NAVSEA question regarding O/B pressure for the EXO-26 FFM with 600 ft umbilical. General info for MK 3 LWDS console and umbilicals with any FFM.

Analyzed influence of raising O/B pressure from 135 psig to 165 psig for 300 and 600 ft umbilicals. Used 582 data points in the NEDU database, a 3D curve fitting software to find equations defining the performance of the MK 3 console and the MK 21 helmet.

From that analysis came the following recommendations:

1) When using 600 ft umbilicals with the MK 2 LWDS, set console O/B pressure to 165 psig, regardless of depth or diver work rate.

2) W/ 300 ft umb., set O/B to 165 psig deeper than 130 fsw. For hard work, 165 psig O/B can help shallower than 130 fsw.

26 Apr 99: MK 3 LWDS Certification Suggestions

1) Estimated friction factor for Gates hoses based on published pressure drops and Lew Nuckols' *PipeFlow* program.

2) Found from A&T letter (12 Aug 96) console pressure drop for 65 acfm flow rate.

3) Use *PipeFlow* to find equivalent length of console internal piping and valves (50 ft of 3/8 in hose).

4) Use above information to calculate console pressure drop for 52 acfm flow.
   2 divers at 2.2 acfm in demand mode at 190 fsw (2 x 15 = 30 scfm)
   1 diver at 3.2 acfm steady flow (22 scfm at 190 fsw)

5) Combined console and umbilical pressure drop:
   a. 15 scfm per umbilical – OK (sideblock pressure = 116 psi O/B)
   b. 22 scfm per umbilical – not OK (sideblock pressure too low at 190 fsw)

26 Apr 99: draft NEDU Technical Letter submitted to NAVSEA for comment: Support for a 2.2 acfm certification flow rate. Recommend MK 3 LWDS be certified to support one diver with a full open steady flow valve at 190 fsw (with at least 3.2 acfm flow), and two divers on demand mode at 190 fsw.

APPENDIX B. Flow Meter Descriptions

The following descriptions of flow meters were copied from the web site of the Iowa Energy Center of Iowa State University. The web site address is: http://www.energy.iastate.edu/DDC_online/

Methods for Measuring Flow

Flow rate is typically obtained by measuring a velocity of a fluid in a duct or pipe and multiplying this velocity by the known cross sectional area (at the point of measurement) of that duct or pipe. Common methods for measuring airflow include hot wire anemometers, differential pressure measurement systems, and vortex shedding sensors. Common methods used to measure liquid flow include differential pressure measurement systems, vortex shedding sensors, positive displacement flow sensors, turbine based flow sensors, magnetic flow sensors, ultrasonic flow sensors and 'target' flow sensors.

Hot Wire Anemometers

'Hot Wire" or thermal anemometers operate on the principle that the amount of heat removed from a heated temperature sensor by a flowing fluid can be related to the velocity of that fluid. Most sensors of this type are constructed with a second, unheated temperature sensor to compensate the instrument for variations in the temperature of the air. Hot wire sensors are available as single point instruments for test purposes, or in multi-point arrays for fixed installation. Hot wire type sensors are better at low airflow measurements than differential pressure types, and are commonly applied to air velocities from 50 to 12,000 feet per minute.

Differential Pressure Measurement Systems

Differential pressure measurement technologies can be applied to both airflow and liquid flow measurements. Sensor manufacturers offer a wide variety of application specific sensors used for airflow and pressure measurements, as well as 'wet-to-wet' differential pressure sensors used for liquid measurements. Both lines offer a wide variety of ranges.

A concentric orifice plate is the simplest and least expensive of the differential pressure type meters. The orifice plate constricts the flow of a fluid to produce a differential pressure across the plate (Figure B-1). The result is a high pressure upstream and a low pressure downstream that is proportional to the square of the flow velocity. An orifice plate usually produces a greater overall pressure loss than other flow elements. An advantage of this device is that cost does not increase significantly with pipe size.

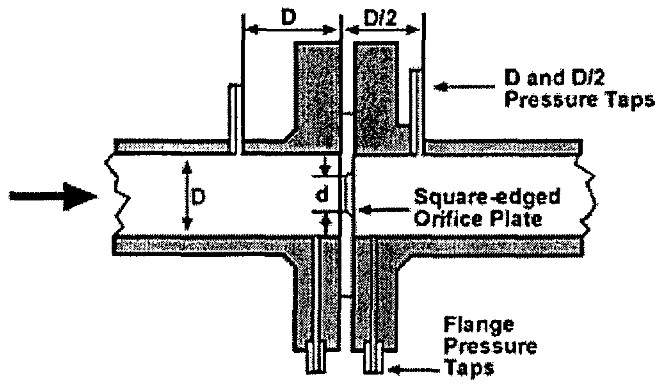

Figure B-1. The concentric orifice.

Venturi tubes exhibit a very low pressure loss compared to other differential pressure meters, but they are also the largest and most costly. They operate by gradually narrowing the diameter of the pipe, and measuring the resultant drop in pressure (see Figure B-2). An expanding section of the meter then returns the flow to very near its original pressure. As with the orifice plate, the differential pressure measurement is converted into a corresponding flow rate. Venturi tube applications are generally restricted to those requiring a low pressure drop and a high accuracy reading. They are widely used in large diameter pipes.

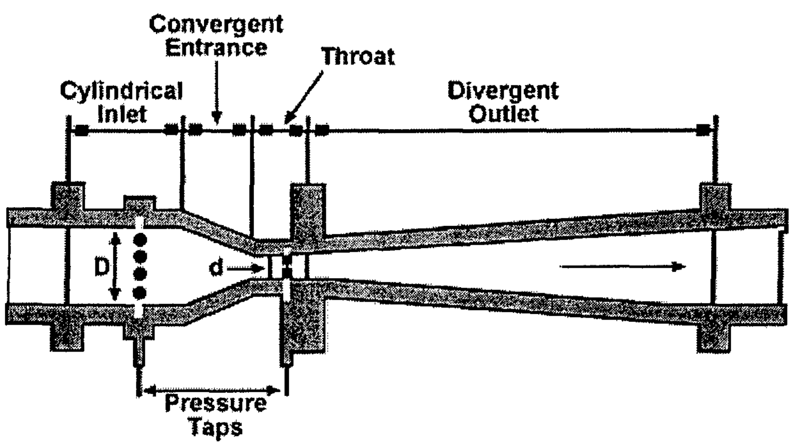

Figure B-2. The Venturi tube.

Flow nozzles may be thought of as a variation on the Venturi tube. The nozzle opening is an elliptical restriction in the flow but with no outlet area for pressure recovery (Figure B-3). Pressure taps are located approximately 1/2 pipe diameter downstream and 1 pipe diameter upstream. The flow nozzle is a high velocity flow meter used where turbulence is high (Reynolds numbers above 50,000) such as in steam flow at high temperatures.

The pressure drop of a flow nozzle falls between that of the Venturi tube and the orifice plate (30 to 95 percent).

The turndown (ratio of the full range of the instrument to the minimum measurable flow) of differential pressure devices is generally limited to 4:1. With the use of a low range transmitter in addition to a high range transmitter or a high turndown transmitter and appropriate signal processing, this can sometimes be extended to as great as 16:1 or more. Permanent pressure loss and associated energy cost is often a major concern in the selection of orifices, flow nozzles, and venturis. In general, for a given installation, the permanent pressure loss will be highest with an orifice type device, and lowest with a Venturi. Benefits of differential pressure instruments are their relatively low cost, simplicity, and proven performance.

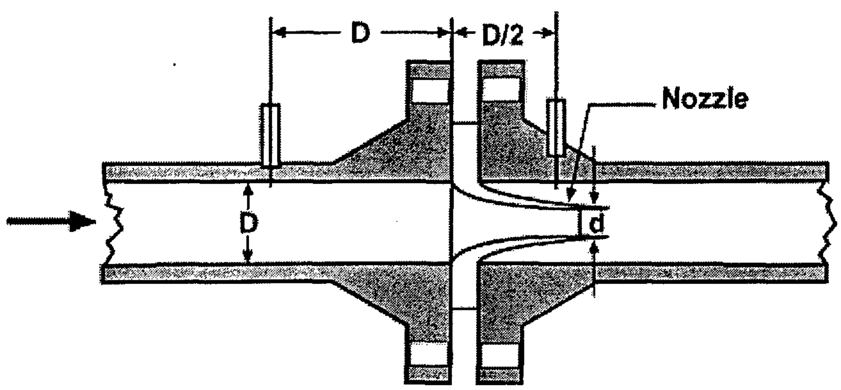

Figure B-3. The flow nozzle.

Vortex Shedding Sensors

Vortex shedding flow meters operate on the principle (Von Karman) that when a fluid flows around an obstruction in the flow stream, vortices are shed from alternating sides of the obstruction in a repeating and continuous fashion. The frequency at which the shedding alternates is proportional to the velocity of the flowing fluid. Single sensors are applied to small ducts, and arrays of vortex shedding sensors are applied to larger ducts, similar to the other types of airflow measuring instruments. Vortex shedding airflow sensors are commonly applied to air velocities in the range of 350 to 6000 feet per minute.

Vortex flow meters provide a highly accurate flow measurement when operated within the appropriate range of flow. Vortex meters are commonly applied where high quality water, gas and steam flow measurement is desired. Performance of up to 30:1 turndown on liquids and 20:1 on gases and steam with 1-2 percent accuracy is available. Turndowns are based on liquid velocities through the meter of up to 25 feet per second for liquids, 15,000 feet per minute for steam and gases. Actual turndown may be less depending on design velocity limitations.

Positive Displacement Flow Sensors

Positive displacement meters are used where high accuracy at high turndown is required and reasonable to high permanent pressure loss will not result in excessive energy consumption. Applications include water metering such as for potable water service, cooling tower and boiler make-up, and hydronic system make-up. Positive displacement meters are also used for fuel metering for both liquid and gaseous fuels. Common types of positive displacement flow meters include lobed and gear type meters, nutating disk meters, and oscillating piston type meters. These meters are typically constructed of metals such as brass, bronze, cast and ductile iron, but may be constructed of engineered plastic, depending on service.

Due to the close tolerance required between moving parts of positive displacement flow meters, they are sometimes subject to mechanical problems resulting from debris or suspended solids in the measured flow stream. Positive displacement meters are available with flow indicators and totalizers that can be read manually. When used with Direct Digital Controller (DDC) systems, the basic meter output is usually a pulse that occurs at whatever time interval is required for a fixed volume of fluid to pass through the meter. Pulses may be accepted directly by the DDC controller and converted to flow rate, or total volume points, or a separate pulse to analog transducer may be used. Positive displacement flow meters are one of the more costly meter types available.

Turbine Based Flow Sensors

Turbine and propeller type meters operate on the principle that fluid flowing through the turbine or propeller will induce a rotational speed that can be related to the fluid velocity. Turbine and propeller type flow meters are available in full bore, line mounted versions and insertion types where only a portion of the flow being measured passes over the rotating element. Full bore turbine and propeller meters generally offer medium to high accuracy and turndown capability at reasonable permanent pressure loss. With electronic linearization, turndowns to 100:1 with 0.1% linearity are available. Insertion types of turbine and propeller meters represent a compromise in performance to reduce cost. Typical performance is 1 percent accuracy at 30:1 turndown. Turbine flow meters are commonly used where good accuracy is required for critical flow control or measurement for energy computations. Insertion types are used for less critical applications. Insertion types are often easier to maintain and inspect because they can be removed for inspection and repair without disturbing the main piping. Some types can be installed through hot tapping equipment and do not require draining of the associated piping for removal and inspection.

For airflow measurements, differential pressure flow devices in common use in HVAC systems include Pitot tubes (Figure B-4) and various types of proprietary velocity pressure sensing tubes, grids, and other arrays. All of these sensing elements are combined with a low differential pressure transmitter to produce a signal that is proportional to the square root of the fluid velocity. For example, when using a Pitot-static tube, this signal can be related to the flow according to the following equation:

$$Velocity = C \cdot \sqrt{\frac{2 \cdot VP \cdot g_c}{\rho}}$$

Where:

Velocity = Velocity (ft/min)
VP = velocity pressure (in w.c.)
p = density of air (lbm/ft2)
gc = gravitational constant (32.174 lbm× ft/lbf×s2)
C = unit conversion factor (136.8)

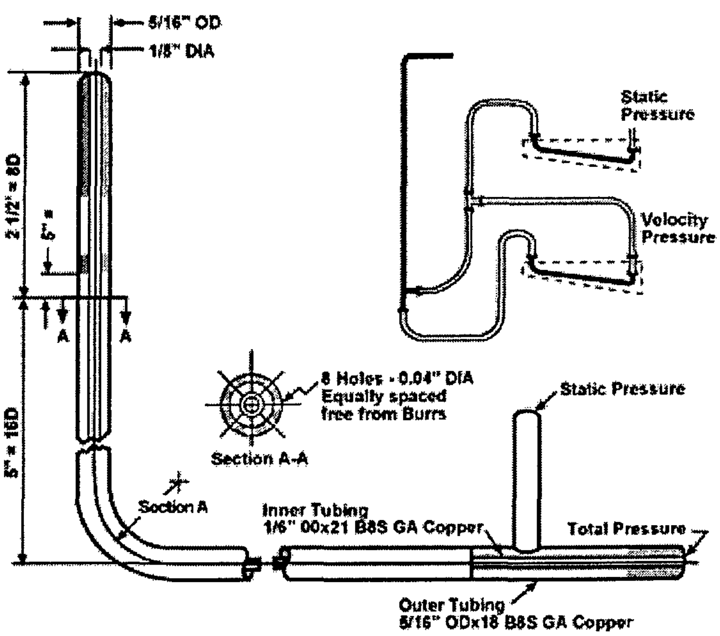

Figure B-4. The Pitot tube.

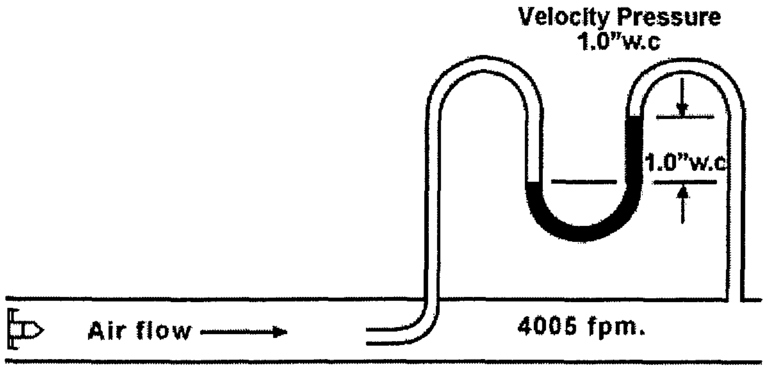

Figure B-5. Velocity pressure with U-tube manometer.

Figure B-5 depicts an example of a velocity pressure measurement with a U tube manometer and Figure B-6 depicts an example of the relationship between velocity pressure (VP), static pressure, and total pressure.

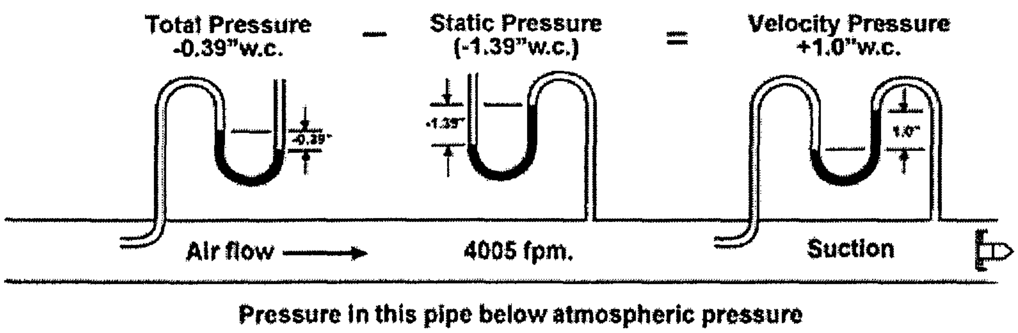

Figure B-6. Relationship between total, static, and velocity pressure.

Target Flow Sensors

A target meter consists of a disc or a "target" which is centered in a pipe (see Figure B-7). The target surface is positioned at a right angle to the fluid flow. A direct measurement of the fluid flow rate results from the force of the fluid acting against the target. Useful for dirty or corrosive fluids, target meters require no external connections, seals, or purge systems.

Target flow meters are commonly used to for liquid flow measurement and less commonly applied to steam and gas flow. Target Meters offer turndowns up to 20:1 with accuracy around 1%.

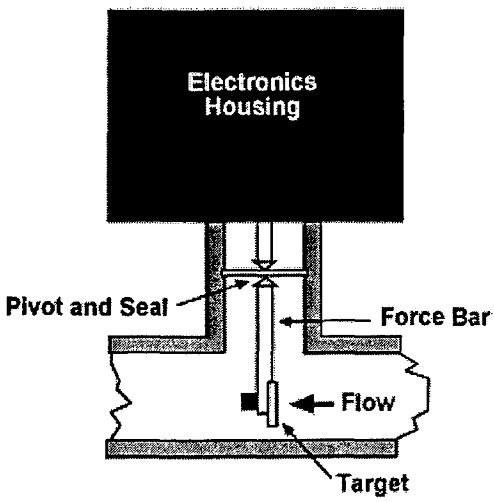

Figure B-7. The target flow meter.

Installation

All airflow sensors work best in sections of ducts that have uniform, fully developed flow. All airflow sensing devices should be installed in accordance with the manufacturers recommended straight runs of upstream and downstream duct in order to provide reliable measurement. A number of manufacturers offer flow straightening elements that can be installed upstream of the sensing array to improve undesirable flow conditions. These should be considered when conditions do not permit installation with the required straight runs of duct upstream and downstream from the sensor.

As with airflow, all liquid flow sensors work best when fully developed, uniform flow is measured. To attain fully developed, uniform flow sensors should be installed in accordance with the manufacturers recommended straight runs of upstream and downstream pipe in order to provide the most reliable measurements.
With most liquid flows measured for HVAC applications, density changes with pressure and temperature are relatively small and most often ignored due to their insignificant

effect on flow measurements. When measuring the flow of steam or fuel gases, unless temperature and pressure are constant, ignoring the effect density changes with varying temperature and pressure will often result in significant or gross errors. For this reason, it is common to measure the temperature and pressure, in addition to the flow, and electronically correct the result for the fluid density. This correction may be done using an integral or remote microprocessor based "flow computer" or it may be made in the DDC controller with suitable programming.

Appendix C. Ambiguities in the Electrical-Pneumatic Flow Analogy

Resistance is a measure of a ratio: the ratio of pressure drop across a resistor and the flow passing through the resistor. In electrical analogs, flow is electrical current, expressed in amps or some fraction of amps. In gas flow through pipes, flow is ambiguous. There are many flows, and therefore many resistances that might be of interest. For example, flow might be expressed as volumetric flow under standard conditions; e.g. in units of scfm, or flow might be expressed under ambient conditions (acfm). In physiological measurements of gas exchange it is the volumetric flow at ambient conditions that determines the effectiveness of gas exchange. Mass flow rate might also be useful, as might gas velocity.

The Right Resistance

In the following example, we gain an appreciation for the type of flow that is most relevant in expressing the resistance of pipes or hoses in diving applications. Assume we have a 3/8 in. umbilical that is 658-feet long. One end is open to a 265-psia pressure source. The other end is exposed to either 14.7 psia (1 atm abs) or 100 psia (the approximate pressure at 190 fsw). Internal umbilical roughness (e) equals 0.0001 ft We will consider the largest pressure drop condition first.

Using *PipeFlow* we find that for a pressure drop (ΔP) of 250 psi, volumetric flow rate ($\dot{V}$) at standard conditions is 32.9 scfm. Mass flow rate ($\dot{M}$) is 2.5 lbm/min. The volumetric flow under ambient conditions varies along the length of the pipe, due to the pressure gradient. At the high pressure inlet, flow is 1.8 acfm (Table 1). At the outlet, flow is 32.8 acfm. Gas velocity varies from 39.5 ft/sec at the inlet to 712.6 ft/sec (Mach 0.63) at the outlet, due to gas expansion within the confines of the hose. If we express resistance as the ratio of pressure drop along the hose (250 psi) and volumetric flow rate at the outlet under ambient conditions, $R_{\dot{V}out}$ = 7.6 psi/acfm. The resistance is the same ($R_{\dot{V}}$ = 7.6 psi/scfm) when expressed as flow under standard conditions.

Table 1. Outlet pressure of 14.7 psia.

	P (psi)	$\dot{V}$ **acfm**	**v (ft/sec)**
Inlet	265.0	1.82	39.53
Outlet	14.7	32.80	712.64
	ΔP (psi)	$\bar{\dot{V}}$ **acfm**	$\bar{v}$ **ft/s**
	250.3	17.31	376.09
$R_{\dot{V}}$ (psi/scfm)	7.6		
$R_{\bar{\dot{V}}}$ (psi/acfm)	14.5		
$R_{\dot{V}out}$ (psi/acfm)	7.6		
$R_{\dot{M}}$ (psi/lbm/min)	100.5		
R_{v} (psi/ft/s)	0.7		

L = 658 ft
e = 0.0001 ft
$\dot{V}$ = 32.92 scfm
$\dot{M}$ = 2.49 lbm/min
T = 68° F

However, when the outlet pressure is 100 psi, the ΔP is only 165 psi. Volumetric flow rate under standard conditions drops slightly, from 32.9 scfm to 30.5 scfm (Table 2). Mass flow rate drops from 2.5 lbm/min to 2.3 lbm/min. Flow rate at the inlet is also slightly reduced from 1.82 to 1.69 acfm. However, flow rate at the outlet is markedly reduced, from 32.8 acfm to 4.47 acfm. Gas velocity also drops from Mach 0.62 to Mach 0.086 (97.0 ft/sec). Resistance based on volumetric flow at the outlet ($R_{\dot{V}out}$) is 5.1 psi/acfm under ambient conditions, whereas $R_{\dot{V}}$ is 5.4 psi/scfm, reduced from 7.6 psi/scfm in the previous example.

Table 2. Outlet pressure of 100 psia.

	P (psi)	$\dot{V}$ **acfm**	**v (ft/sec)**
Inlet	265.00	1.69	36.6
Outlet	100	4.47	97.0
	ΔP (psi)	$\bar{\dot{V}}$ **acfm**	$\bar{v}$ **ft/s**
	165.0	3.08	66.80
$R_{\dot{V}}$ (psi/scfm)	5.4		
$R_{\bar{\dot{V}}}$ (psi/acfm)	53.6		
$R_{\dot{V}out}$ (psi/acfm)	5.1		
$R_{\dot{M}}$ (psi/lbm/min)	72.7		
R_{v} (psi/ft/s)	2.5		

L = 658 ft
e = 0.0001 ft
$\dot{V}$ = 30.49 scfm
$\dot{M}$ = 2.27 lbm/min
T = 68° F

Intuitively, the flow resistance of the 658-foot umbilical should be only moderately affected by the down stream pressure. By reducing the pressure drop (driving pressure) across the 658-feet of umbilical from 250 psi to 165 psi (34% reduction), both mass flow rate and volumetric flow under standard conditions is reduced 7% - 9%. The resistance under standard conditions drops almost proportionately with the pressure drop; i.e., about 29%. If we think of mass flow rate or volumetric flow under standard conditions as analogous to current, and if resistance is directly proportional to current, then we would expect resistance to drop in the electrical analogy whenever current (flow) is reduced.

On the other hand, the other potential measures of resistance ($R_{\bar{\dot{V}}}$ (psi/acfm) and R_{v} (psi/ft/s)) actually increased almost 4 times when outlet pressure increased from 14.7 psia to 100 psia. This result does not make sense from an electrical analog perspective. Since our goal was to use both pneumatic and electrical analogs to analyze the MK3 system for this report, it made sense to use pneumatic resistance values based on standard flows.