AMCA International

Fan Power in 90.1 & IECC

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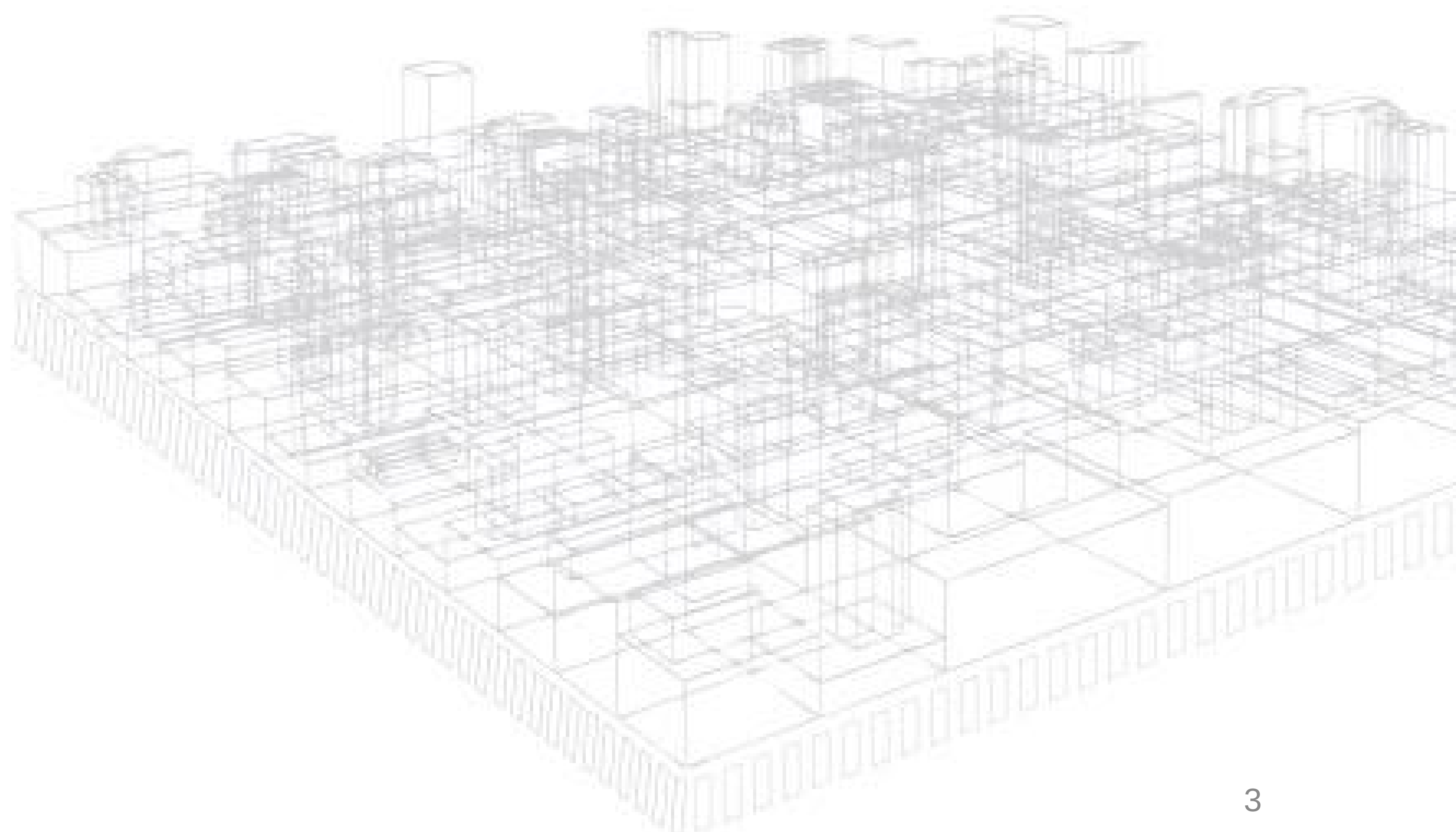

Learning Objectives

- Understand the Fan Power Requirements of 90.1
- Understand differences between the Fan Power Requirements of 90.1 and IECC
- Learn techniques to verify compliance prior to design completion (no redesign)
- Understand how FEG and FEI affect compliance now and may in future editions of the codes
  - Session 2, Track 2 will delve deeper into this topic
- Learn tips to help comply early in your projects without alienating your architect or client

Jeff Boldt, PE, FASHRAE, LEED® AP, HBDP

Principal – **IMEG Corp**

- Member ASHRAE SSPC-90.1
  - Chair Hydronics, Elevator, Booster pumps & HW Working Groups
- Member ASHRAE **SSPC-189.1**
  - Chair Acoustics Discussion Group
- Author Advanced Energy Design Guide for Large Hospitals, & AEDG for Small Healthcare Facilities
- TC 3.6 Water Treatment
- TC 5.2 – Duct Systems
- GPC-36 – Advanced Control Sequences
- SSPC-215P – Operating System Duct Leakage
- PE - M and FP
- Jeff.G.Boldt@IMEGcorp.com

Search for "ASHRAE Seminar Recordings" or "Jeff Boldt Nerd"

Engineering nerd watches 1,772 hours of ASHRAE ...

www.youtube.com/watch?v=woWi792Vw6I

Jan 20, 2014 - Uploaded by KJWW

In a fantastic display of nerdiness, Jeff Boldt, Director of Engineering for KJWW Engineering Consultants ...

STANDARD

ANSI/ASHRAE/IES Standard 90.1-2016
(Supersedes ANSI/ASHRAE/IES Standard 90.1-2013)
Includes ANSI/ASHRAE/IES addenda listed in Appendix H

Energy Standard for Buildings Except Low-Rise Residential Buildings (I-P Edition)

See Appendix H for approval dates by the ASHRAE Standards Committee, the ASHRAE Board of Directors, the IES Board of Directors, and the American National Standards Institute.

This Standard is under continuous maintenance by a Standing Standard Project Committee (SSPC) for which the Standards Committee has established a documented program for regular publication of addenda or revisions, including procedures for timely, documented, consensus action on requests for change to any part of the Standard. The change submittal form, instructions, and deadlines may be obtained in electronic form from the ASHRAE website (www.ashrae.org) or in paper form from the Senior Manager of Standards. The latest edition of an ASHRAE Standard may be purchased from the ASHRAE website (www.ashrae.org) or from ASHRAE Customer Service, 1791 Tullie Circle, NE, Atlanta, GA 30329-2305. E-mail: orders@ashrae.org. Fax: 678-539-2129. Telephone: 404-636-8400 (worldwide), or toll free 1-800-527-4723 (for orders in US and Canada). For reprint permission, go to www.ashrae.org/permissions.

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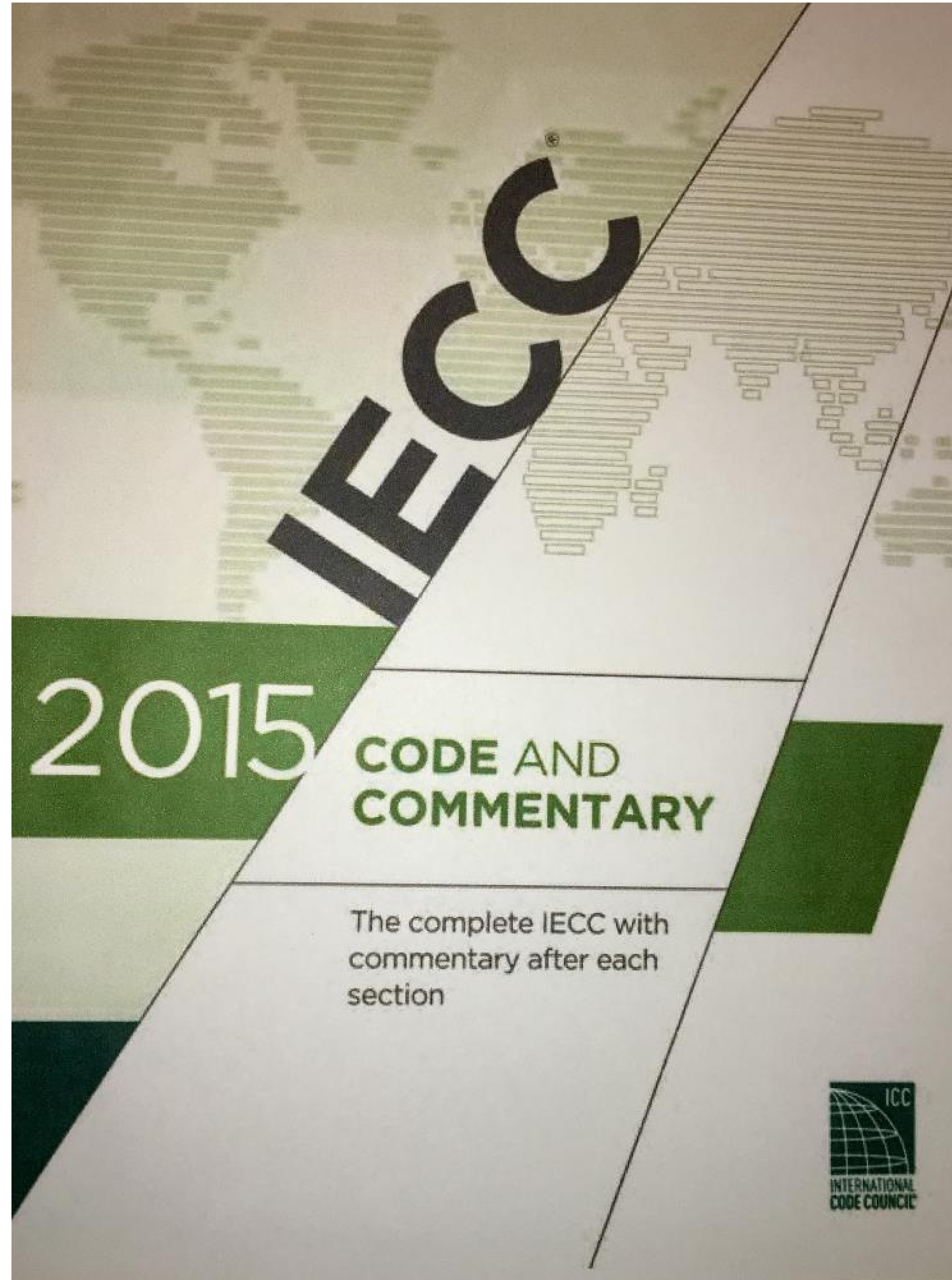


What's Required by 90.1? Prescriptive Path Only.

- 2007 & 2010 & 2013 & 2016

  6.5.3 Air System Design and Control. Each HVAC system having a total *fan system motor nameplate hp* exceeding 5 hp shall meet the provisions of Sections 6.5.3.1 through 6.5.3.2. (.4 in 2010) (.5 in 2013) (merged into 6.5.3.1 in 2016)

  6.5.3.1 Fan System Power Limitation (Power & Efficiency in 2013)

  6.5.3.1.1 Each HVAC system **at fan system design conditions** shall not exceed the allowable *fan system motor nameplate hp* (Option 1) **or fan system bhp** (Option 2) as shown in Table 6.5.3.1.1A. This includes supply fans, return/relief fans, exhaust fans, and fan-powered terminal units associated with systems providing heating or cooling capability.

  (2010+) Single zone variable-air-volume systems shall comply with the constant volume fan power limitation.

Not All Fans Count

- **6.5.3.1.1** Each … **<u>at fan system design conditions</u>** ….
  - –Relief fans often don't count and can be <u>very</u> inefficient
    - Something I'd like to work on in 90.1
  - –Cabinet heater fans
    - Shaded pole, PSC, ECM
  - –Parallel (heating) fan powered boxes don't count
    - Some controversy about this
    - IMO - takeaway from RP-1292 should be that parallel boxes should have better backdraft dampers, not that series boxes save energy
    - Some believe the message is that series boxes are more efficient
    - Also IMO – series may be good on critical branches – depends on time of day and hours/week in critical branch

What's Required by 90.1?

- 2007, 2010, 2013, & 2016

Exceptions:

a. Hospital, vivarium, and laboratory systems that utilize flow control devices on exhaust and/or return to maintain space pressure relationships necessary for occupant health and safety or environmental control may use variable-volume fan power limitation.

b. Individual exhaust fans with motor nameplate ≤1 hp.

c. Fans exhausting air from fume hoods. Note: If this exception is taken, no related exhaust side credits shall be taken from Table 6.5.3.1.1B and the Fume Hood Exhaust Exception Deduction must be taken from Table 6.5.3.1.1B. (deleted in 2010, 13, & 16)

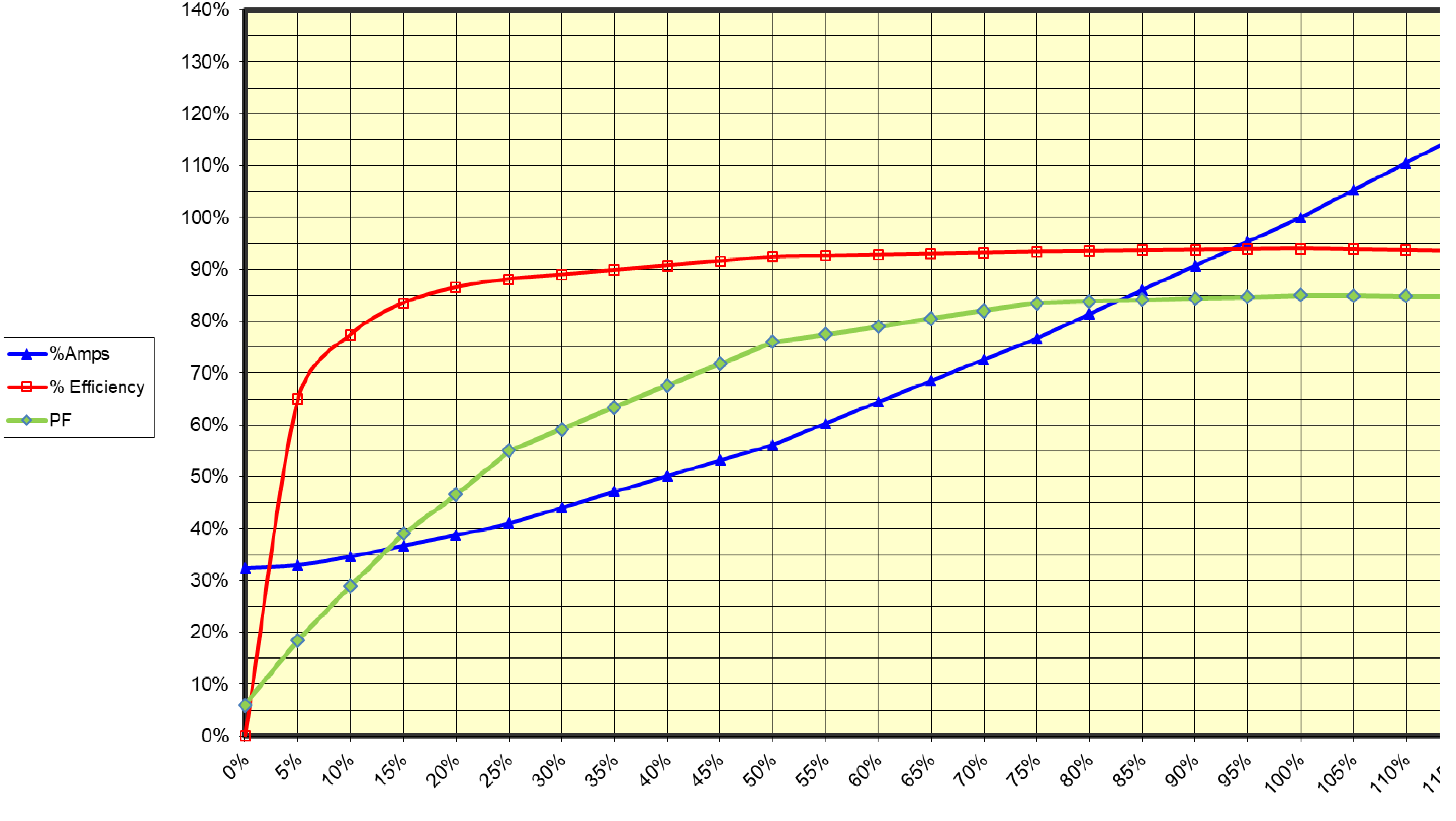
What's Required by 90.1?

- **6.5.3.1.2 Motor Nameplate Horsepower.** For each fan, the selected fan motor shall be no larger than the **first available motor size greater than the bhp**. The fan bhp must be indicated on the design documents to allow for compliance verification by the code official (2016 *building official*). (2019 Update?)

- Which bhp? Fans are rated without including belt losses.

- **Amps ≠ Energy!**

3-Phase Motor Performance – Very Different for Single-Phase!



Fan Motor Sizing

Exceptions:

1. For fans less than 6 bhp, where the first available motor larger than the brake horsepower has a name-plate rating within 50% of the bhp, the next larger nameplate motor size may be selected.

2. For fans 6 bhp and larger, where the first available motor larger than the bhp has a nameplate rating within 30% of the bhp, the next larger nameplate motor size may be selected.

3. Systems complying with Section 6.5.3.1.1, Option 1. (2013)

4. Fans with motor nameplate horsepower of less than 1 hp. (2016)

Is the BHP Method a Loophole?

- Normally the method we use
- We don't spec over 90% motor loading
  - Possibly 90.1 violation for us for a 9.5 BHP load
- Do some practitioners or companies oversize motors because it's needed for the actual BHP?
  - I don't know whether this is widespread

What's Required by 90.1?

TABLE 6.5.3.1.1A Fan Power Limitation[a]

	Limit	Constant Volume	Variable Volume
Option 1: Fan System Motor Nameplate hp	Allowable Nameplate Motor hp	hp ≤ $CFM_S \cdot 0.0011$	hp ≤ $CFM_S \cdot 0.0015$
Option 2: Fan System bhp	Allowable Fan System bhp	bhp ≤ $CFM_S \cdot 0.00094 + A$	bhp ≤ $CFM_S \cdot 0.0013 + A$

[a] where

CFM_S = the maximum design supply airflow rate to conditioned spaces served by the system in cubic feet per minute
hp = the maximum combined motor nameplate horsepower
bhp = the maximum combined fan brake horsepower
A = sum of ($PD \times CFM_D$/4131)

where

PD = each applicable pressure drop adjustment from Table 6.5.3.1.1B in in. w.c.
CFM_D = the design airflow through each applicable device from Table 6.5.3.1.1B in cubic feet per minute

TABLE 6.5.3.1.1B Fan Power Limitation Pressure Drop Adjustment

Device	Adjustment
Credits	
Fully ducted return and/or exhaust air systems	0.5 in. w.c.
Return and/or exhaust airflow control devices	0.5 in. w.c.
Exhaust filters, scrubbers, or other exhaust treatment	The pressure drop of device calculated at fan system design condition
Particulate Filtration Credit: MERV 9 through 12	0.5 in. w.c.
Particulate Filtration Credit: MERV 13 through 15	0.9 in. w.c.
Particulate Filtration Credit: MERV 16 and greater and electronically enhanced filters	Pressure drop calculated at 2× clean filter pressure drop at fan system design condition
Carbon and other gas-phase air cleaners	Clean filter pressure drop at fan system design condition
Heat recovery device	Pressure drop of device at fan system design condition
Evaporative humidifier/cooler in series with another cooling coil	Pressure drop of device at fan system design condition
Sound Attenuation Section	0.15 in. w.c.
Deductions	
Fume Hood Exhaust Exception (required if 6.5.3.1.1 Exception [c] is taken)	–1.0 in. w.c.

Table 6.5.3.1-2 Fan Power Limitation Pressure Drop Adjustment Device A - 2016

	Device	Adjustment
	Credits	
2016Δ	Return or exhaust *systems* required by code or accreditation standards to be fully ducted, or *systems* required to maintain air pressure differentials between adjacent rooms.	0.5 in. of water (2.15 in. of water for laboratory and vivarium *systems*)
	Return and/or exhaust airflow *control devices*	0.5 in. of water
	Exhaust filters, scrubbers, or other exhaust treatment	The pressure drop of device calculated at *fan system design condition*
	Particulate Filtration Credit: MERV 9 through 12	0.5 in. of water
	Particulate Filtration Credit: MERV 13 through 15	0.9 in. of water
	Particulate Filtration Credit: MERV 16 and greater and electronically enhanced filters	Pressure drop calculated at 2× clean filter pressure drop at *fan system design condition*
	Carbon and other gas-phase air cleaners	Clean filter pressure drop at *fan system design condition*
2010>>	Biosafety cabinet	Pressure drop of device at *fan system design condition*
2010Δ	*Energy* recovery device, other than coil runaround loop	For each airstream [(2.2 × *Enthalpy Recovery Ratio*) – 0.5] in. of water
	Coil runaround loop	0.6 in. of water for each airstream
	Evaporative humidifier/cooler in series with another cooling coil	Pressure drop of device at *fan system design condition*
2013Δ	Sound attenuation section (fans serving *spaces* with design background noise goals below NC35)	0.15 in. of water
2007-1”	Exhaust *system* serving fume hoods	0.35 in. of water
	Laboratory and vivarium exhaust *systems* in high-rise *buildings*	0.25 in. of water/100 ftof vertical duct exceeding 75 ft
	Deductions	
2016	*Systems* without central cooling device	–0.6 in. of water
“	*Systems* without central heating device	–0.3 in. of water
“	*Systems* with central *electric resistance* heat	–0.2 in. of water

IECC-2015 is Similar to 90.1-2016

- Because fan power addenda occurred early in the cycle they were in time for the ICC hearings
- In the early 2000’s there was no fan power limit in the IECC

FEG & FEI

- Fan Efficiency Grade (FEG)
  - Entered 90.1 in 2013
  - Required FEG ≥67 & within 15 points of maximum efficiency per AMCA 205 at design condition
  - Exceptions:
    - <5HP – single fans or arrays
    - Equipment covered by 6.4.1.1
    - Equipment where fans are included in energy ratings
    - PRVs
    - Fans outside scope of AMCA 205
    - Emergency fans

FEI

- Fan Efficiency Index (FEI)
  - –Improvement over FEG
  - –FEG almost surely being removed from 90.1-2019
  - –Debate about replacing it with FEI
    - I'm betting YES
    - Limit on the worst efficiency fans
      - –Relief fans
      - –Return fans in packaged equipment
      - –Systems that have no trouble meeting fan power limits

How to Comply?

- Rule of Thumb
  - –5" SP supply fan should use ~1BHP/1,000 cfm

SD/DD Measures

- A method is provided to estimate where the project stands
- Much cheaper to get it right now than at CD stage!

Step 1 – Airflow and Equipment

- Estimate cfm
  - Often 0.8-1.2 cfm/SF
  - Healthcare ~1.2-1.5 cfm/SF
- More AHU's serving smaller areas with shorter duct runs = easier to make comply with fan power limits
- Coordinate with Architect

Step 2 – Determine AHU Configuration

- Project type often determines the AHU configuration
  - E.g. Blow-thru in healthcare possible power savings but nearly also always = wet final filters
- Variable volume allows ~30% more motor horsepower
- Include all sections and components of the AHU
- Determine AHU internal ΔP
  - Typically AHU ΔP is ~ 50% of total system ΔP
  - Often cheaper to increase AHU size than to increase duct sizes
  - Low ΔP filters can help
  - Can label when to change filters
    - Rule of thumb – lowest LCC is at twice clean pressure drop

Step 3 – Determine Allowable System TSP

- Table 6.5.3.1.1A provides the equation for allowable brake HP for variable volume systems
  - $BHp \leq CFM_S \times 0.0013 + A$
- Substituting and solving for TSP gives
  - $TSP \leq \frac{0.0013 \times \eta_{EFF}}{0.000157} + \frac{A \times \eta_{EFF}}{CFM_S \times 0.000157}$
- Calculate A from Table 6.5.3.1.1B
  - $\mathrm{A} = \sum(PD \times \frac{CFM_D}{4131})$

Step 3 – Continued - Reminder

	Device	Adjustment
	Credits	
2016Δ	Return or exhaust *systems* required by code or accreditation standards to be fully ducted, or *systems* required to maintain air pressure differentials between adjacent rooms.	0.5 in. of water (2.15 in. of water for laboratory and vivarium *systems*)
	Return and/or exhaust airflow *control devices*	0.5 in. of water
	Exhaust filters, scrubbers, or other exhaust treatment	The pressure drop of device calculated at *fan system design condition*
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2016	*Systems* without central cooling device	–0.6 in. of water
“	*Systems* without central heating device	–0.3 in. of water
“	*Systems* with central *electric resistance heat*	–0.2 in. of water

Step 4 – Determine Total Ductwork ΔP

- Subtract AHU component SP from system TSP
  - If you later fail, resize the AHU
- Total Ductwork ΔP is the total allowable external ΔP associated with total system of supply, return, relief *(if operating at peak cooling load)*, and exhaust ducts associated with the air handling system, including TABs and air devices
- Exhaust fans with nameplate $Hp \leq 1.0$ are exceptions and are not included

Step 5 – Determine Static Pressure Setting

- Static pressure setting upstream of terminal air boxes usually falls between 0.70” and 1.0” w.g.
  - Question balancing reports saying that 1.5” is needed
  - 90.1 now requires resetting this, but that won’t help with the initial compliance documents
- Terminal air boxes with reheat coils must not require more than 0.5” W.C. by IMEG specifications
  - Soap Box: This is mostly coil ΔP, when we do energy recovery chillers we use boxes with larger coil face area, e.g. a 6” damper with the coil & box that goes in a normal 8” box
- Air devices typically require approximately 0.1” w.g.
- Ductwork ΔP usually is low downstream of TABs

Step 6 – Determine Allowable Duct ΔP

- Subtract Static Pressure Setting (TAB and downstream pressure drop found in Step 5) from Total Ductwork ΔP from Step 4.
- This allowable duct ΔP includes supply ductwork upstream of TAB's, return ductwork, and applicable exhaust ductwork (serving fans ≥1hp)
  - Need to fix that exemption in 90.1 IMO

Step 7 – Estimate Actual Duct System ΔP

- Review floor plan and sketch a single-line layout from AHU to TABs and AHU to critical return inlet
- Don't need to include TAB or downstream duct & air devices (already accounted for)
- Sketch single-line ductwork layouts for return and applicable exhaust systems
- Complete preliminary duct pressure drop calculations for each system using spreadsheet or your company's tool

Step 8 – Compare Pressure Drops

- Compare Estimated Duct ΔP to Allowable Duct ΔP
- If Actual ΔP is too large, revise duct design
  - Friction factors, duct sizes, **fittings**, etc.
  - Consider duct routing, number, and size of AHU's (cross sectional area)
- Discuss options for reducing total pressure with Architect/Owner

Step 8 – Maybe Step 1!

- Explain to architect **WHY** your mechanical room should be central to the areas served
  - Shorter ducts are cheaper (pay for the marble entry?)
  - Shorter ducts use less energy
  - Shorter ducts need less fan power and cause less noise
  - Shorter ducts use less volume in the building, so some ceilings can be higher, especially at the perimeter
  - Possibly avoid an energy model that the owner might not pay for
- In my experience, if you have reasons architects give you what is right for the project

Example Calculations – Step 1

- Step 1 – Airflow and Equipment
  - 4-story MOB, 15,000 SF per floor
    - Assume 1.2 cfm/sf
    - CFM = 4 floors x 15,000 SF x 1.2 CFM/SF
    - CFM = 72,000 cfm (estimated)
  - Number of AHU’s
    - Based on building geometry, etc., assume 2 AHU’s
    - CFM = 36,000 cfm per AHU

Example Calc – Step 2

- Step 2 – AHU Configuration
  - VAV w/ RF, Heat Wheel, Blender, HW/CW coils
    - RA damper 0.15”
    - OA damper 0.10”
    - 30% Filter 0.21”
    - Heat Wheel 1.00” (air passes through 2x) Keep ΔP low
    - Air Blender 0.30”
    - HW Coil 0.30”
    - CW Coil 0.60”
    - Total ΔP 2.66” w.g.

Example Calc – Step 3

- Step 3 – Calculate Allowable System TSP
  - Determine exceptions for A
    - Ducted Return 0.5” w.g. (if allowed)
    - Heat Wheel 2.2x%eff - 1.0” = 0.76” w.g.
    - $A = \sum(PD \times \frac{CFM_D}{4131})$
    - A = 0.5 x 36,000 / 4131 + 0.76 x 18,000 / 4131= 7.66
  - Assume 65% efficient fan
    - $TSP \leq \frac{0.0013 \times \eta_{EFF}}{0.000157} + \frac{A \times \eta_{EFF}}{CFM_S \times 0.000157}$
    - TSP = 5.382” + 0.88” = 6.262” w.g.
    - Verify fan efficiency later (higher efficiency is good!)

Example Calc – Step 4

- Step 4 – Calculate Total Ductwork ΔP
  - –Subtract AHU SP from System TSP
    - Duct ΔP =6.262” – 2.66”
    - Duct ΔP =3.602” w.g.
  - –If this is a problem, study duct fitting C-factors!

Example Calc – Step 5

- Step 5 – Determine Static Pressure Setting
  - Calculate ΔP of Downstream Devices
    - Terminal Air Box ΔP 0.5”
    - Assume Ductwork ΔP 0.4”
    - Assume Diffuser ΔP 0.1”
    - Static Pressure Setpoint 1.0” w.c.

Example Calc – Step 6

- Step 6 – Allowable Ductwork ΔP
  - Subtract Static Pressure Setpoint from Allowable Total Ductwork ΔP
  - ΔP = 3.60” – 1.0” = 2.60” w.c.

Example Calk – Step 7

- Step 7 – Actual Ductwork ΔP
  - Sketch one-line ductwork layouts
  - Complete Pressure Drop calc for each duct system
  - Add up Supply, Return, and applicable Exhaust pressure drop calculations
  - Compare Allowable and Actual ΔP

Think About Belts

Drive Efficiency	Per AMCA (1980?)		AMCA 203-1990		Per AMCA 207-17	
Input HP (H_i)	Belt Loss	V-Belt ή	Belt Loss	V-Belt ή	V-Belt Drive	Synchronous Belt
5.00					94.3%	96.7%
0.3	32.0%	68.0%	15.0%	85.0%	86.3%	94.0%
0.4	27.0%	73.0%	13.6%	86.4%	87.4%	94.0%
0.6	19.9%	80.1%	11.2%	88.8%	88.9%	94.0%
0.8	17.0%	83.0%	10.1%	89.9%	89.9%	94.0%
1	15.1%	84.9%	9.5%	90.5%	90.6%	94.0%
2	11.3%	88.7%	7.4%	92.6%	92.5%	94.5%
3	9.6%	90.4%	6.6%	93.4%	93.4%	95.2%
4	8.4%	91.6%	6.1%	93.9%	93.9%	96.0%
6	7.5%	92.5%	5.6%	94.4%	94.5%	97.5%
8	6.6%	93.4%	5.3%	94.7%	94.8%	98.0%
10	6.1%	93.9%	5.1%	94.9%	95.1%	98.0%
20	5.1%	94.9%	4.6%	95.4%	95.5%	98.0%
30	4.5%	95.5%	4.4%	95.6%	95.7%	98.0%
40	4.1%	95.9%	4.3%	95.7%	95.7%	98.0%
60	3.7%	96.3%	4.2%	95.8%	95.8%	98.0%
80	3.4%	96.6%	4.1%	95.9%	95.9%	98.0%
100	3.2%	96.8%	4.1%	96.0%	95.9%	98.0%
200	2.7%	97.3%	4.0%	96.0%	95.9%	98.0%
300	2.5%	97.5%	4.0%	96.0%	96.0%	98.0%
400	2.3%	97.7%	4.0%	96.0%	96.0%	98.0%
600	2.2%	97.8%	4.0%	96.0%	96.0%	98.0%

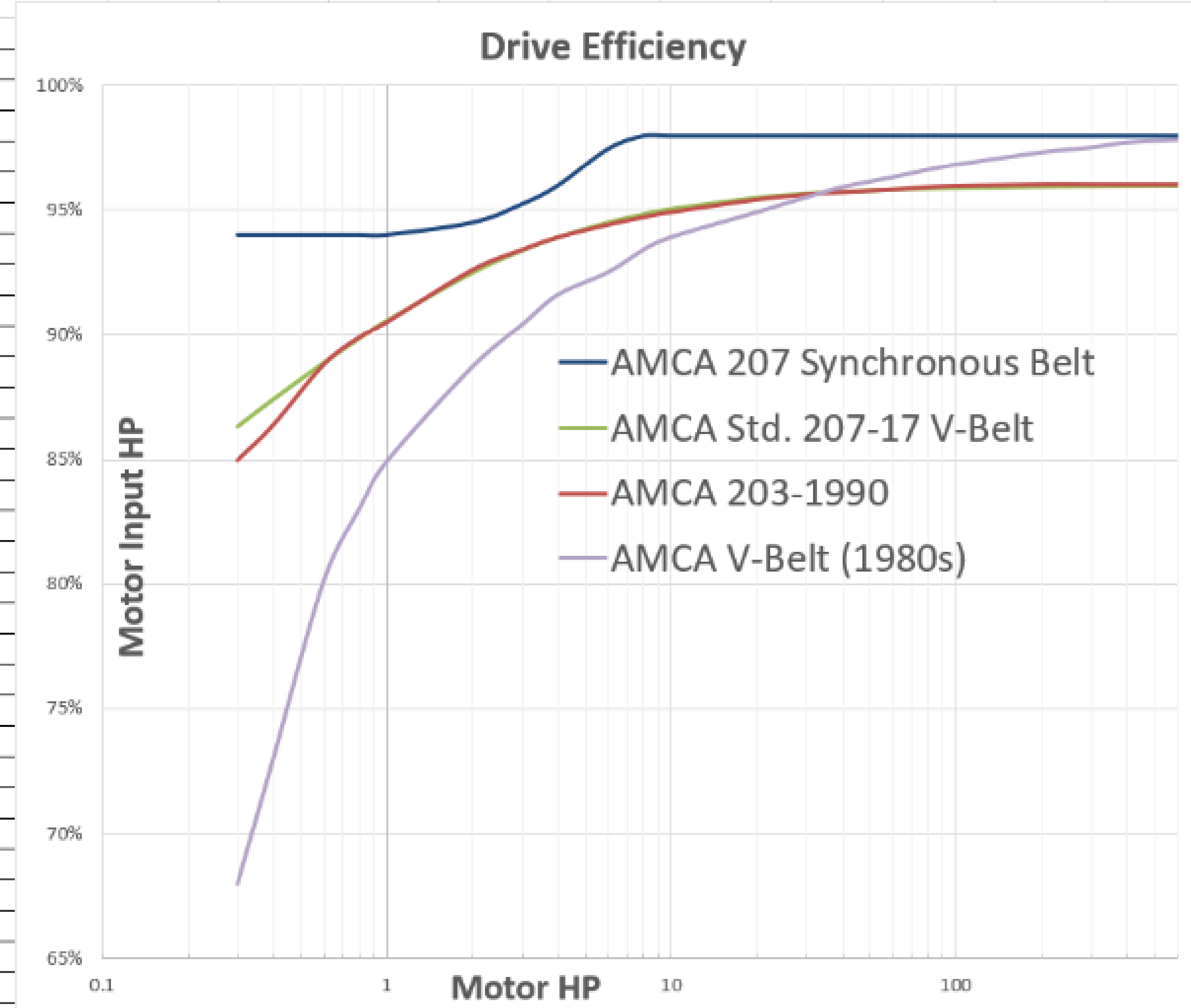
$$\eta_B = 0.96\left(\frac{H_i}{H_i+2.2}\right)^{0.05}$$

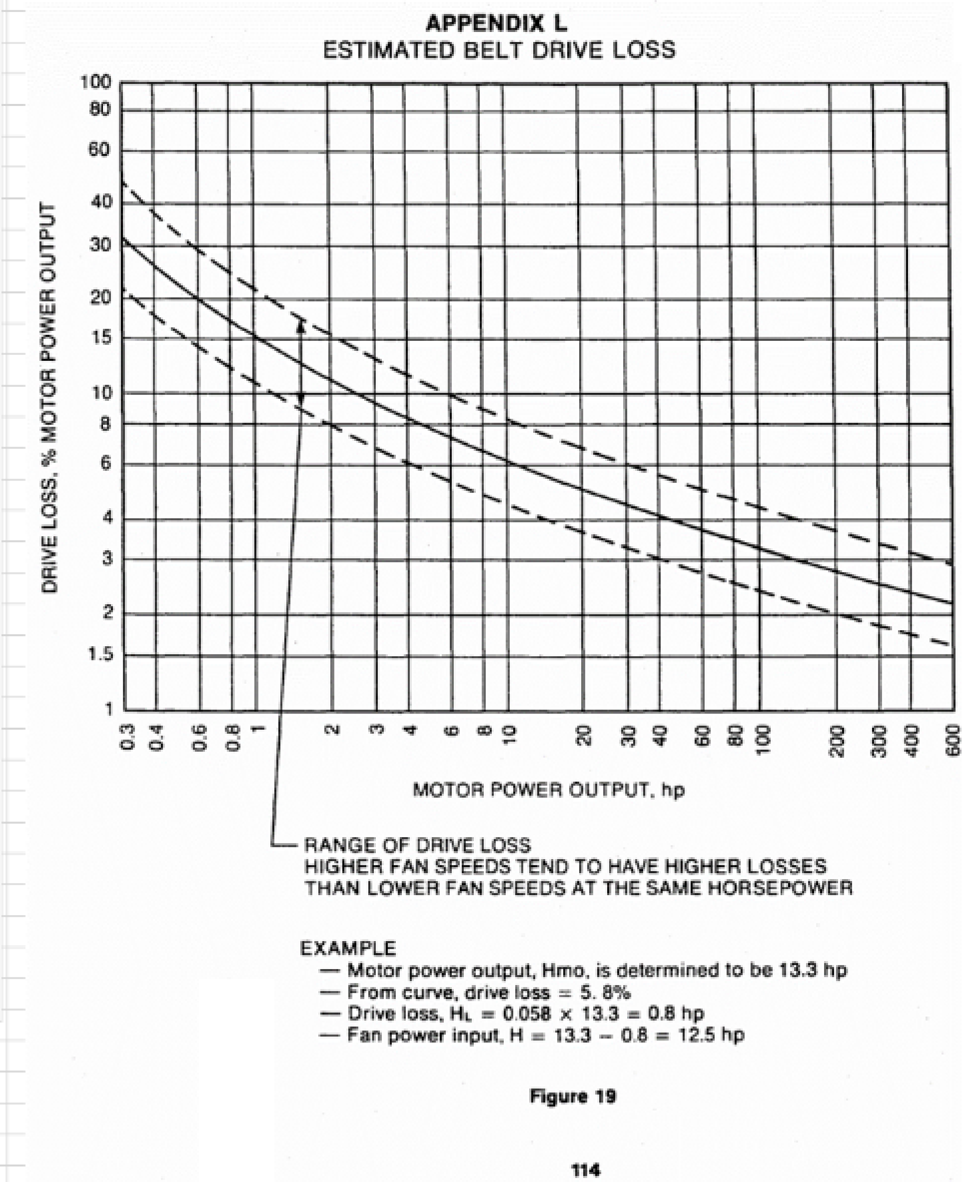
AMCA 207 V-Belt Equ AMCA 207 V-Belt Equation

$$H_i \le 1.34\ hp, \eta_B = 0.94$$
$$1.34\ hp < H_i \le 6.7\ hp, \eta_B = 0.00746\ H_i + 0.93$$
$$H_i > 6.7\ hp, \eta_B = 0.98$$

AMCA 207 Synchrono AMCA 207 Synchronous Belt Equations

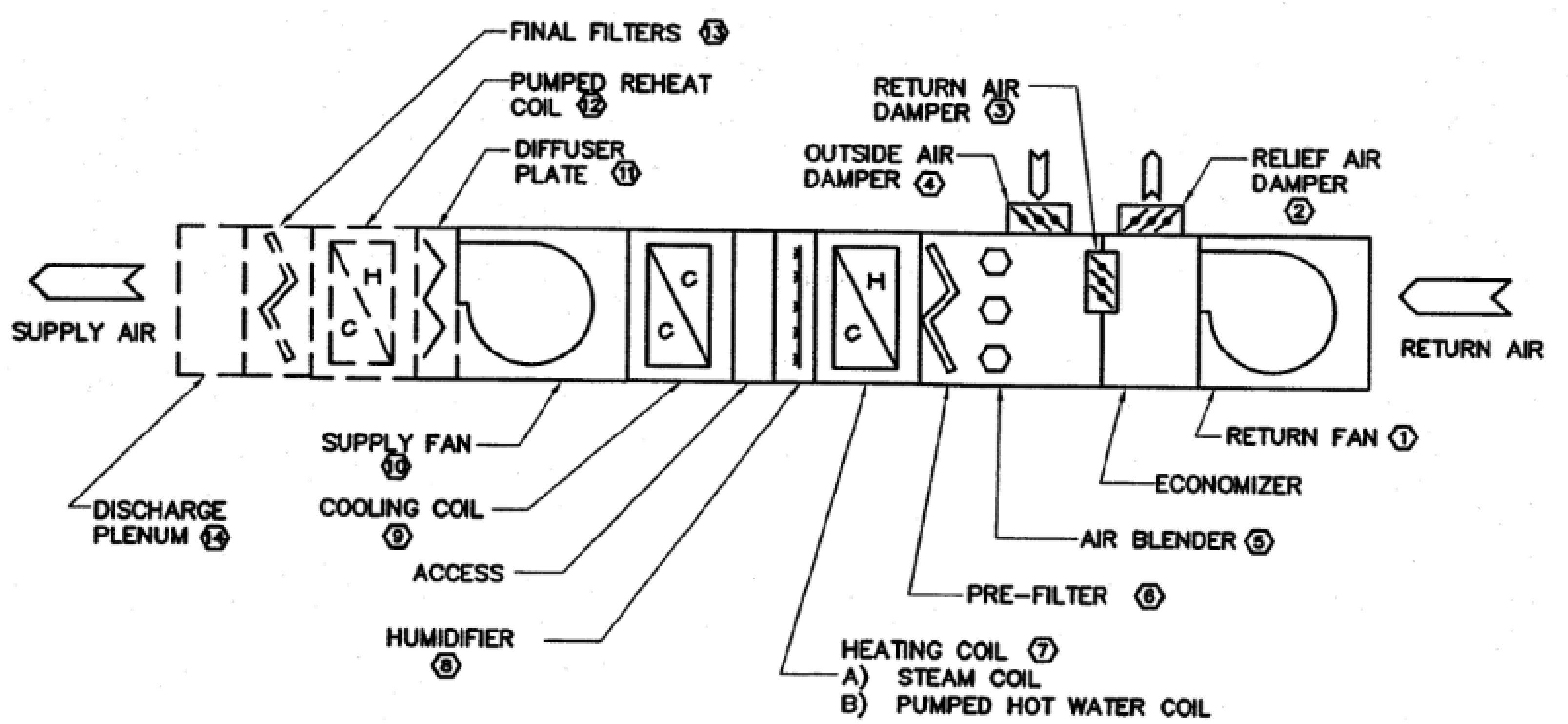
Flexible Coupling Effi Flexible Coupling Efficiency = 98% per AMCA 207-2017





AMCA 203 from ~1980

Basic Workbook – 4 Worksheets



ASHRAE 90.1 - 2007 — AHU CMT, Aug-08

Fan Power Limitation

(Healthcare Application)

X %	ISP For AHU
Y%	ESP for Supply and Return Ductwork
Z%	TSP For General Exhaust
5.5"	TSP (Max TSP to Meet 90.1)

AHU Components

#	Component	Δ P	Credits (ASHRAE 90.1 - 2007)
2	Relief Air Damper/ Louver	0	-0.5 Fully Ducted
3	Return Air Damper	0.15	
4	Outside Air Damper/Louver	0.15	0 Return Air Control Device; -0.5 (Operating Rooms)
5	Air Blender	0.3	
6	Pre-Filter	0.21	
7	Heating Coil	0.3	(Hot water coil)
8	Humidifier	0.08	
9	Cooling Coil	0.6	(0.5" MAY BE ATTAINABLE)
11	Diffuser Plate	0.15	
12	Reheat Coil	0	(No Reheat Coil)
13	Final Filter (not dirty)	0.6	-0.9 95% Filter/MERV 14
		2.54	-1.4

Total ISP = 1.14 " W.C. **X %: ISP FOR AHU**

5.5	TSP (Max TSP to Meet 90.1)
- 1.14	AHU ISP w/ Credits
4.36	**ESP for Supply Fan & Return Fan & GENERAL EXHAUST)**
- **0.75**	**Z%: TSP For General Exhaust**
3.61	**Y%: ESP for Supply and Return Ductwork**

Assume:

2.25 ESP for Supply Fan (TAB = 0.5" DP)

1.36 ESP for Return Fan

(**NOTE:** To be reviewed by Ductwork Committee)

Comments

1. What is the AHU/Coil Velocity, Fins/In, Rows for a cooling coil @ 0.65" W.C.)
2. No reheat coil prior to final filter

B Occupancy Example

- Plenum return
- MERV 7 filtration
- No heat recovery
- Ducted exhaust system is only credit
- Option 1
  - Motor HP ≤ CFMs * 0.0015 = 15 HP of motors
- Option 2
  - BHP ≤CFMs*0.0013 + 1000*0.5/4131 = 13.1 BHP

Exhaust Fan BHP

- Assume
  - EF has 0.75” of static pressure
  - EF requires ~0.3 BHP
  - This leaves 12.8 BHP for the supply fan and the return or exhaust fan

Return Fan BHP

- Plenum return should have ~1” SP
- More if several floors and/or fittings
- Assume 1.5” SP and 9,000 cfm
- If our AHU is a Trane M-30 (yes, this is a 333 fpm coil selection) the most efficient fan pick is the AF, which will require 3.7 BHP

Supply Fan BHP

- We have 9.1 BHP remaining
- The most efficient option is the 22” AF fan, which can deliver 10,000 cfm at up to 4” SP with this amount of input power

Summary – for Office Buildings

- Select efficient fans
- Design for a total supply fan static pressure of ≤4”
- Design for a total return/relief system static of ≤1.5”
- You can probably trade SP between the supply and return fan. The supply fan has more flow, but it is also a more efficient fan
- These rules put you right on the edge of compliance – don’t get caught short

AHU Thoughts

- The AHU should not consume more than 40% of the total SF allowance = 4” * 40% = 1.6”
  - This will typically require the AHU to be selected for ≤400 FPM coil velocity (e.g. a size 30 unit for 10,000 CFM).
  - Blenders will need bypass dampers that open when OA temperature is ≥40°F
  - Supply fans should be oriented to take advantage of their velocity pressure
  - AH Equipment committee will expand improve this list.

Ductwork Thoughts

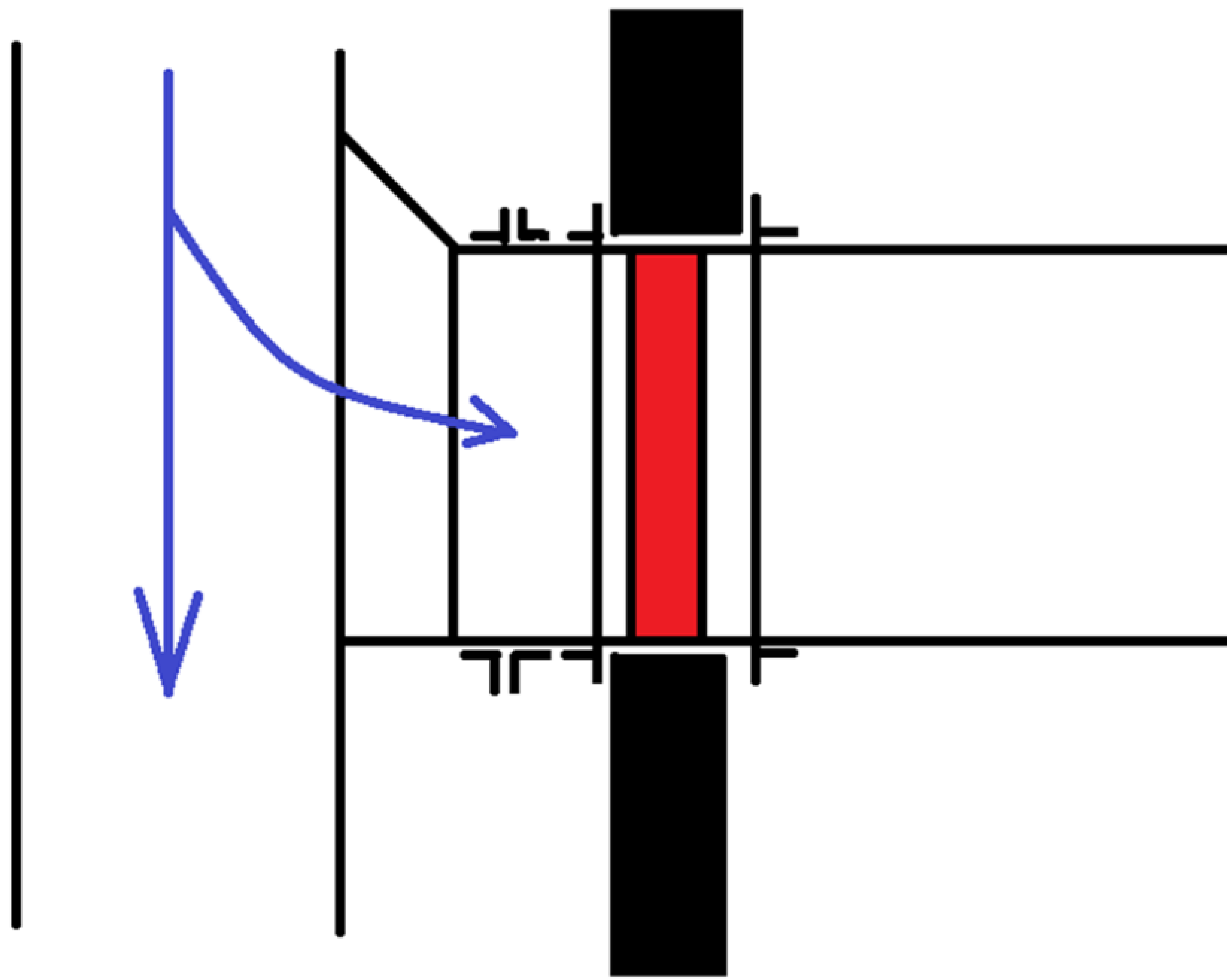
- No more than 5 turns should be needed to any VAV box
- No fittings with C-factors >0.38 should be used in main ducts
- No duct velocity should exceed 2000 FPM unless all fittings will have C-factors <0.20.

Tips to Design Low ΔP Systems

- Good Fittings!
  - Get cozy with the fitting C-factors in the ASHRAE duct fitting database, ASHRAE handbooks, and SMACNA Duct Design Manual
  - Bad fittings are the #1 reason that systems don't comply
  - Use radius elbows when they fit
    - Even R/d=1.0 elbows have less ΔP than turning vane elbows (and duct liner never gets stuck in them)

Good Fittings!

- Design shafts with efficient outlet fittings
  - Fight for space

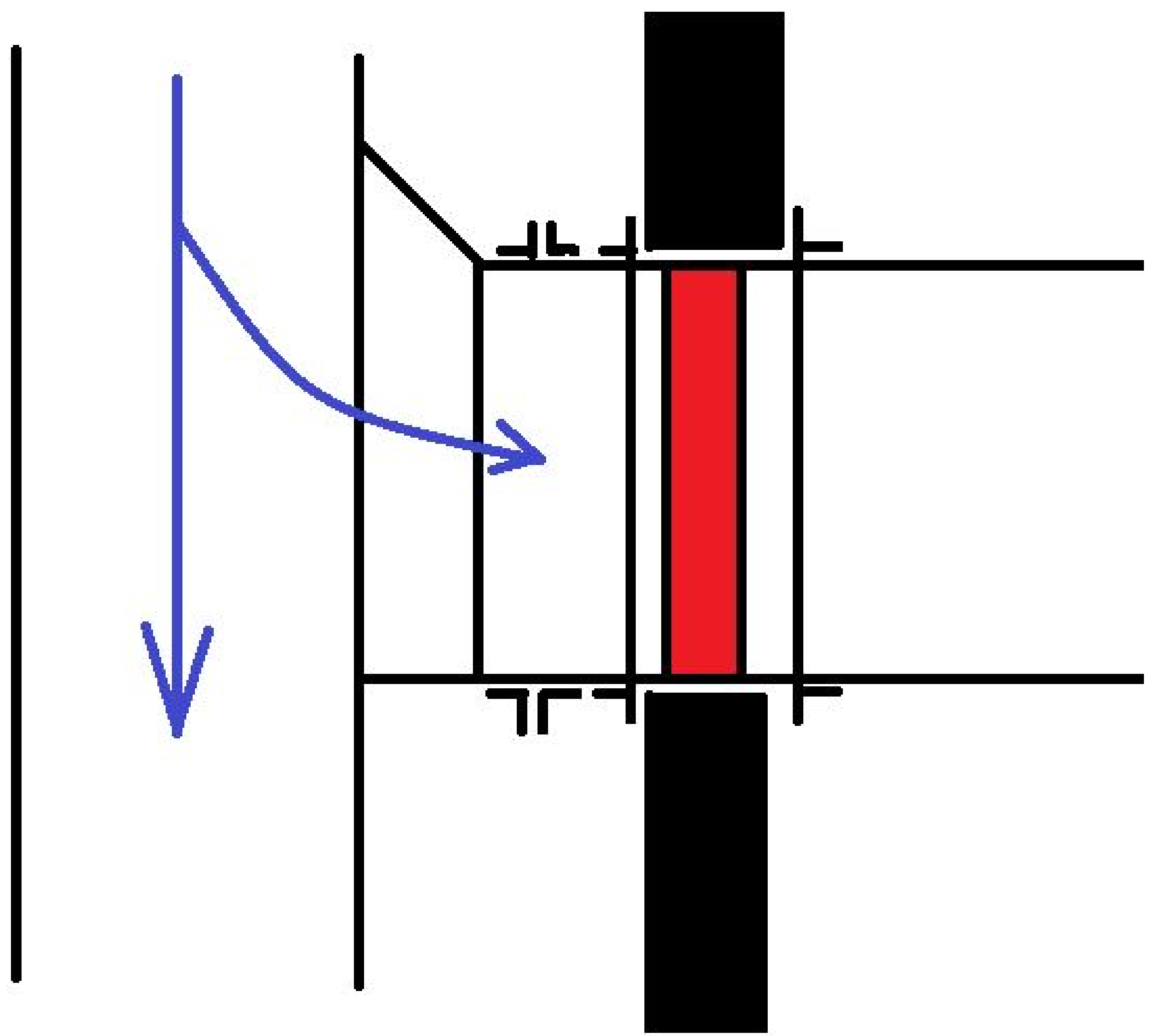


The construction of a 45-degree tap is shown in the Duct Construction Standards – 3rd Edition, Figure 4-6. Imagine the large duct going down a shaft and the 45-degree branch coming off on a floor. The 4-degree offset must extend 4" or ¼ of the depth of the branch on the floor; so probably 4 to 7" plus enough to attach or form a connecting flange.

Fire damper sleeves must stick out of the wall by at least the thickness of the retaining flanges (which varies with duct size) plus enough to make a connection. If flanges are 1.5"x1.5" and a flanged connection system is used, this might be 3" at the very minimum. They are not permitted to extend over 6" from the wall. There are special dampers that can be farther than 6" in listed sleeves. Formerly, I believe NFPA 90A required a minimum of 3" and a maximum of 6", but I'm not sure that was the source. I think today it is the manufacturer Listed minimum and 6" maximum.

So, at a minimum we probably have 4" + 5" = 9" from the inner wall of the shaft to the outer wall of the duct.

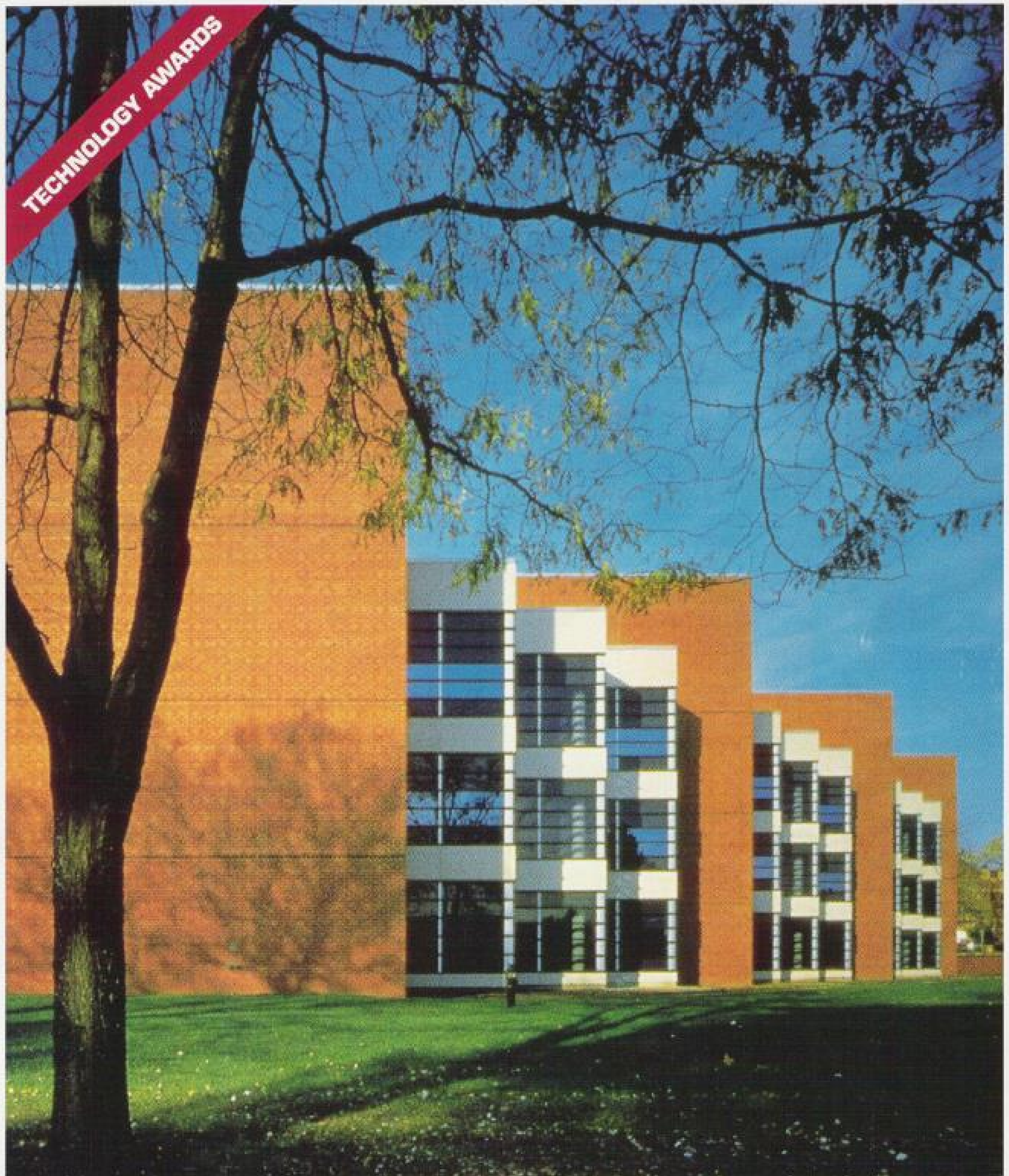
I hope this helps.



ASHRAE JOURNAL

April 1993

The magazine of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.



CONTROL SYSTEMS

THERMAL COMFORT

PROJECT	DATE	BY	PROJECT NO.
Agronomy Fan Power	2013-10-09	JefBol	

AHU-1 (Labs)
120,000 cfm
SF's = 122 BHP Total 150 MHP
RFs = 42 BHP Total 50 MHP
Total = 164 BHP

CC-1. ΔP = 0.70"
HC-1 ΔP = 0.35"

Exhaust Fans:

		Each	Total
53 x 0.45 BHP = 24 BHP ← Don't count		1100 cfm =	58,300 cfm
36 x 0.6 BHP = 22		1500 =	54,000
12 x 1.3 BHP = 16		1250 =	15,000
18 x 0.2 BHP = 4	← Don't count	300 =	5,400
38 ~~56~~ BHP			132,700

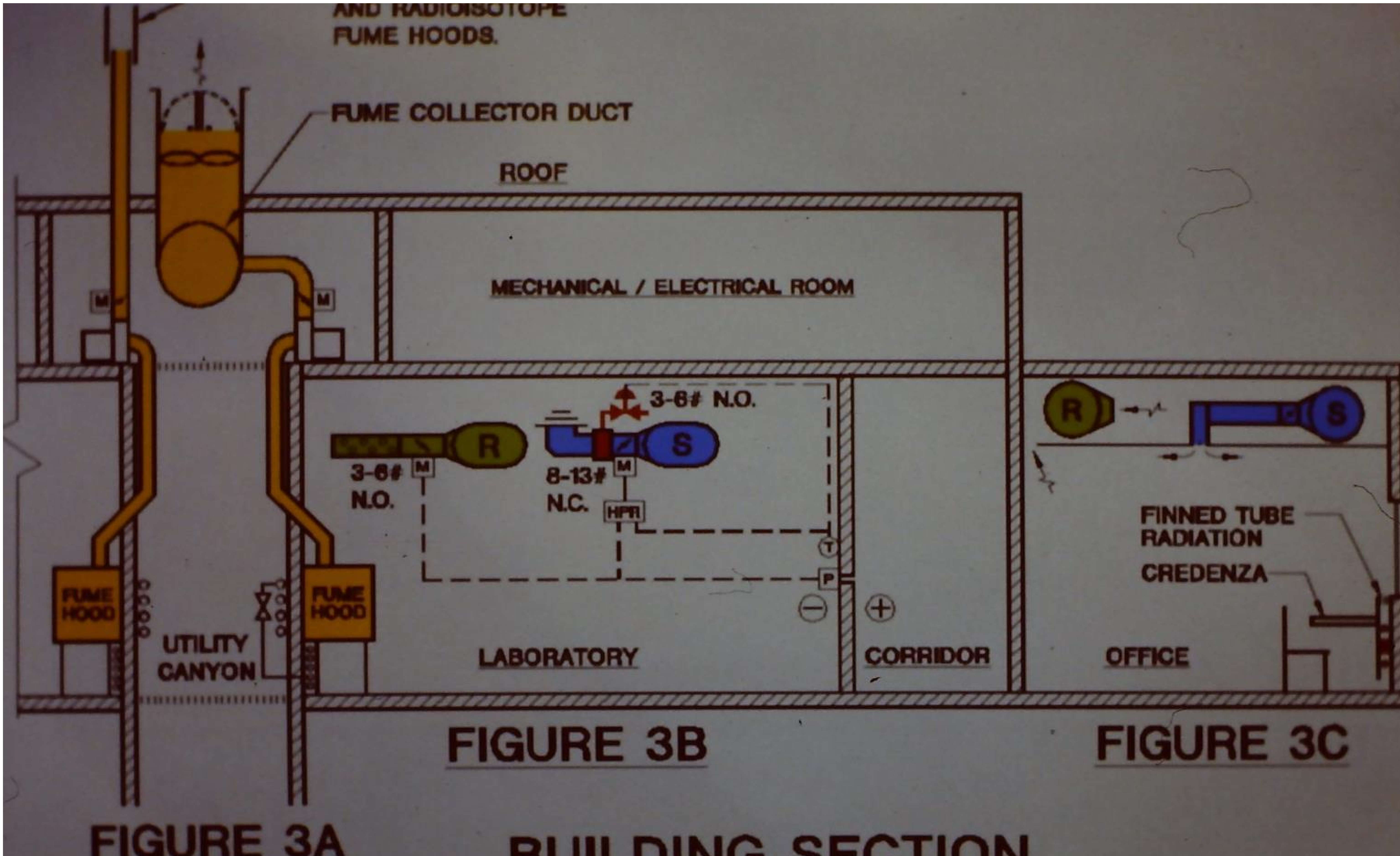
RFs & EFs don't both run full simultaneously

Assume worst case = ~~30~~ 15 BHP EF's + 30 BHP RFs
Total BHP = 122 + 15 + 30 = 167 BHP

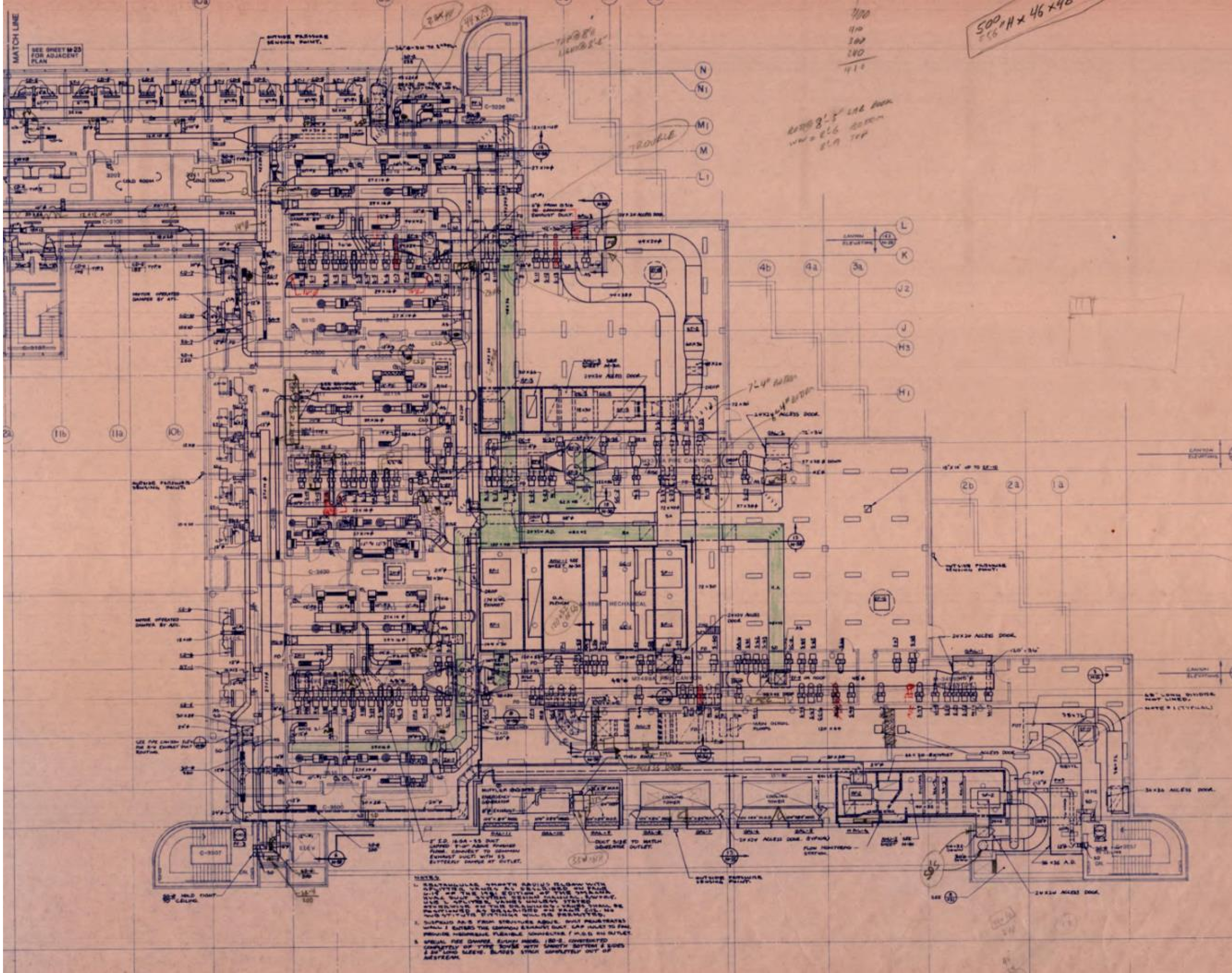
Allowable ↙ .156 11.38 11.25

$$120{,}000 \times 0.0013 + \left(\frac{94{,}000 \times 0.5}{4130}\right) * 2 + \frac{132{,}700 \times 0.35}{4130} = 178.6$$

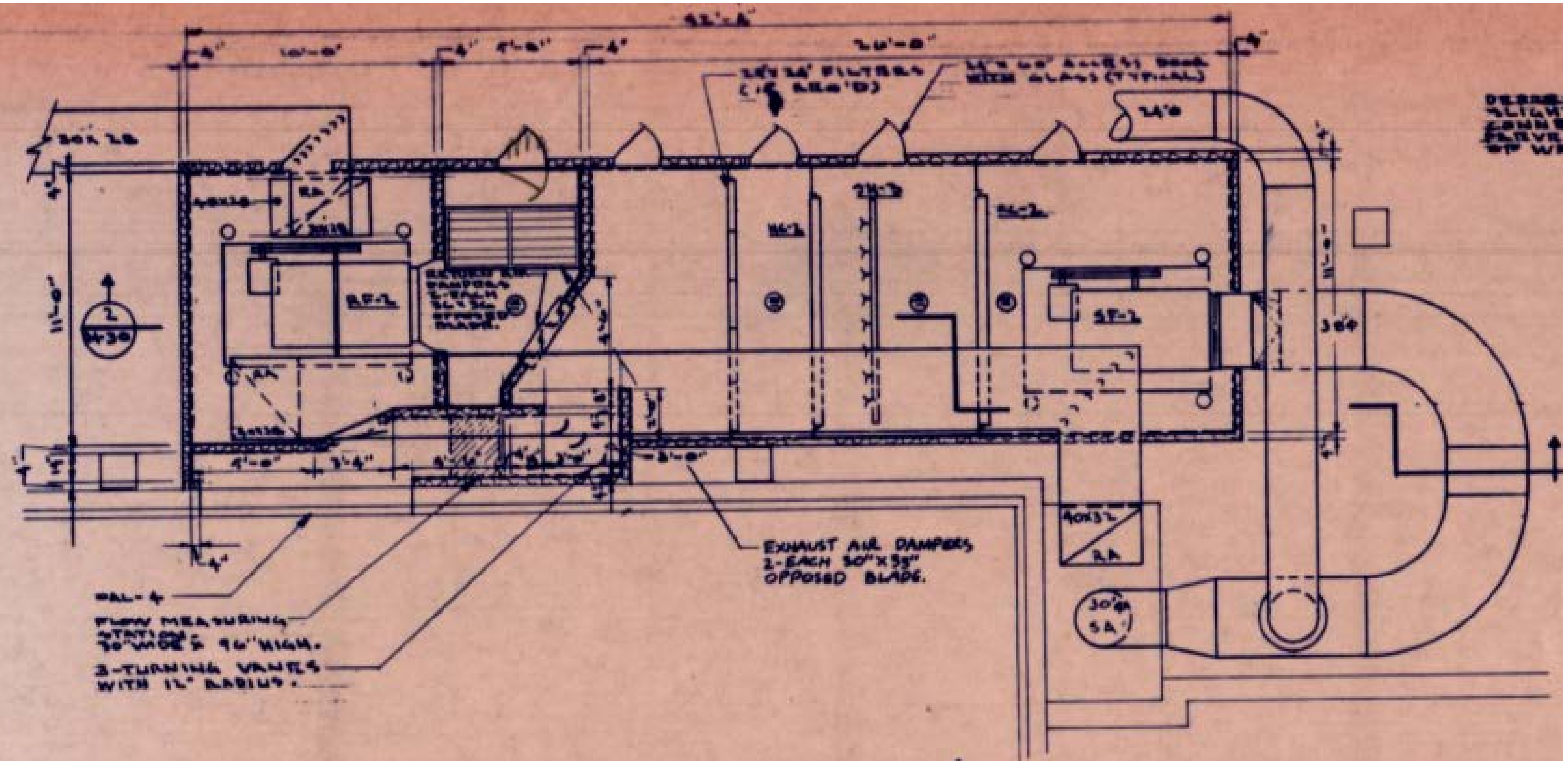
AGRONOMY LAB SYSTEM = 93.5% of allowable BHP even in 90.1-2013



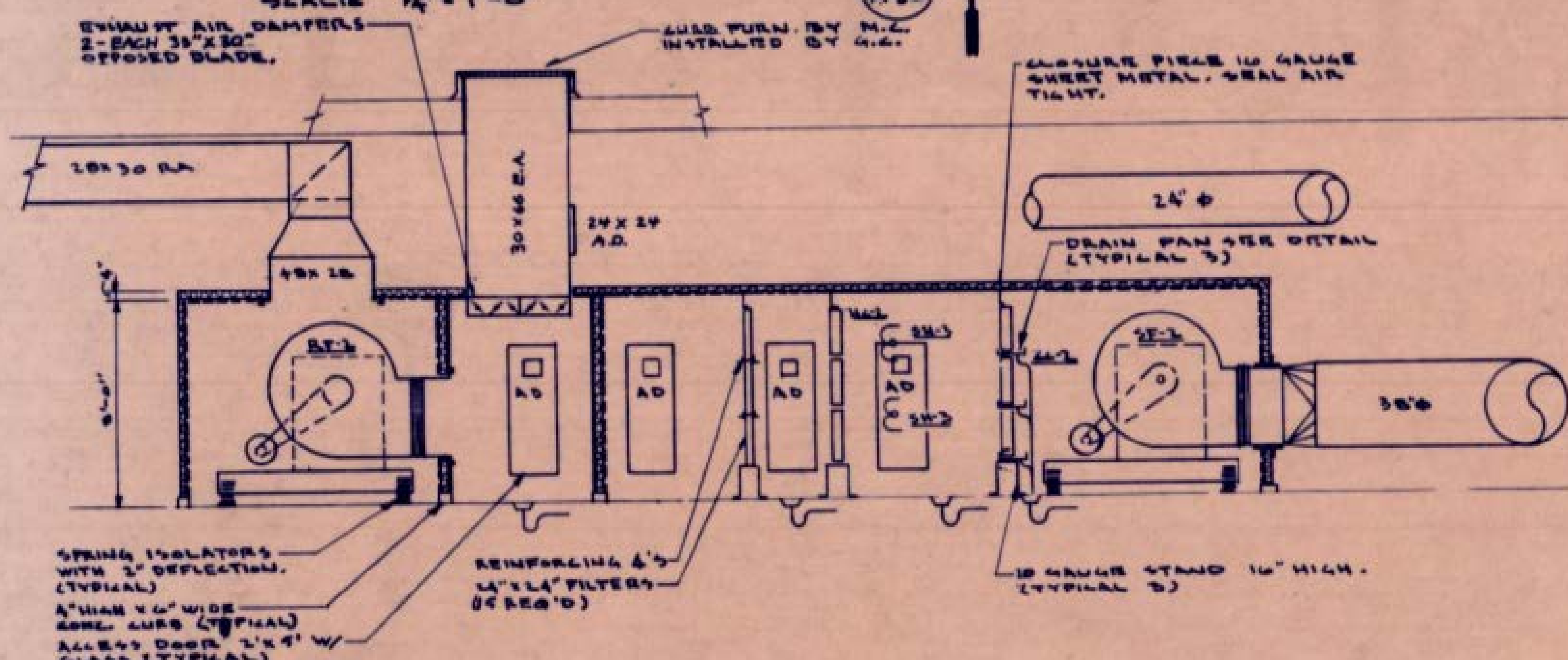
BUILDING SECTION



MATCH LINE
SEE SHEET M-23 FOR ADJACENT PLAN
N
N1
M1
M
L1
L
K
J2
J
H3
H1
4b
4a
3a
2b
2a
1a
11b
11a
10b
TROUBLE
MECHANICAL
NOTES:

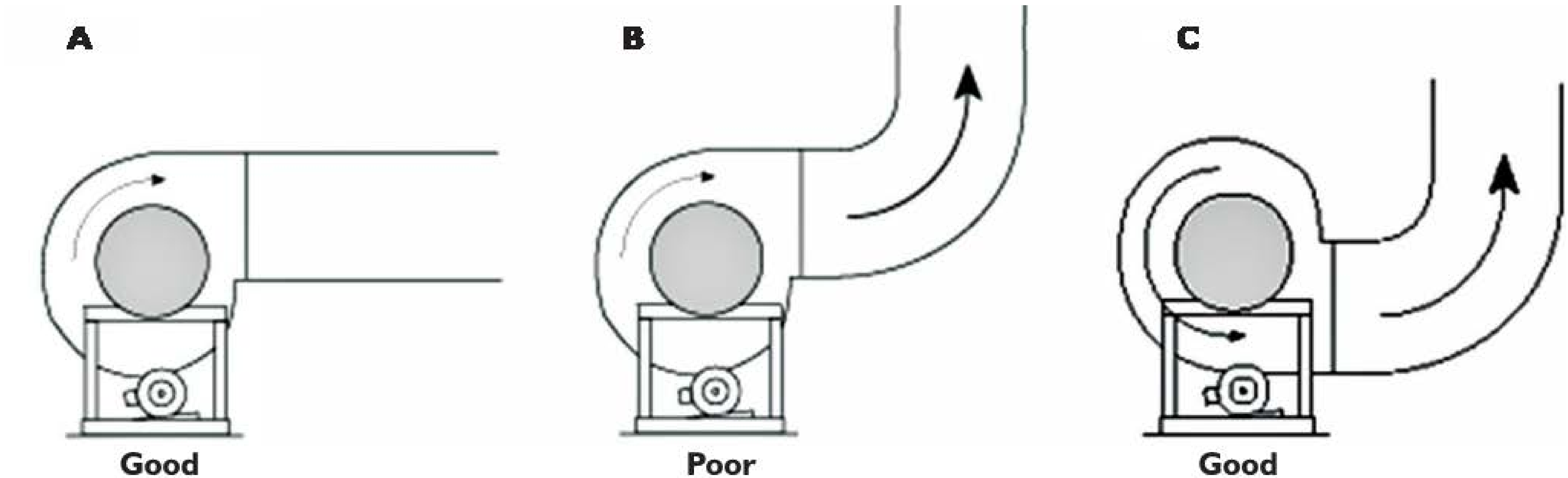


PLAN VIEW - AHU-2
SCALE 1/4" = 1'-0"



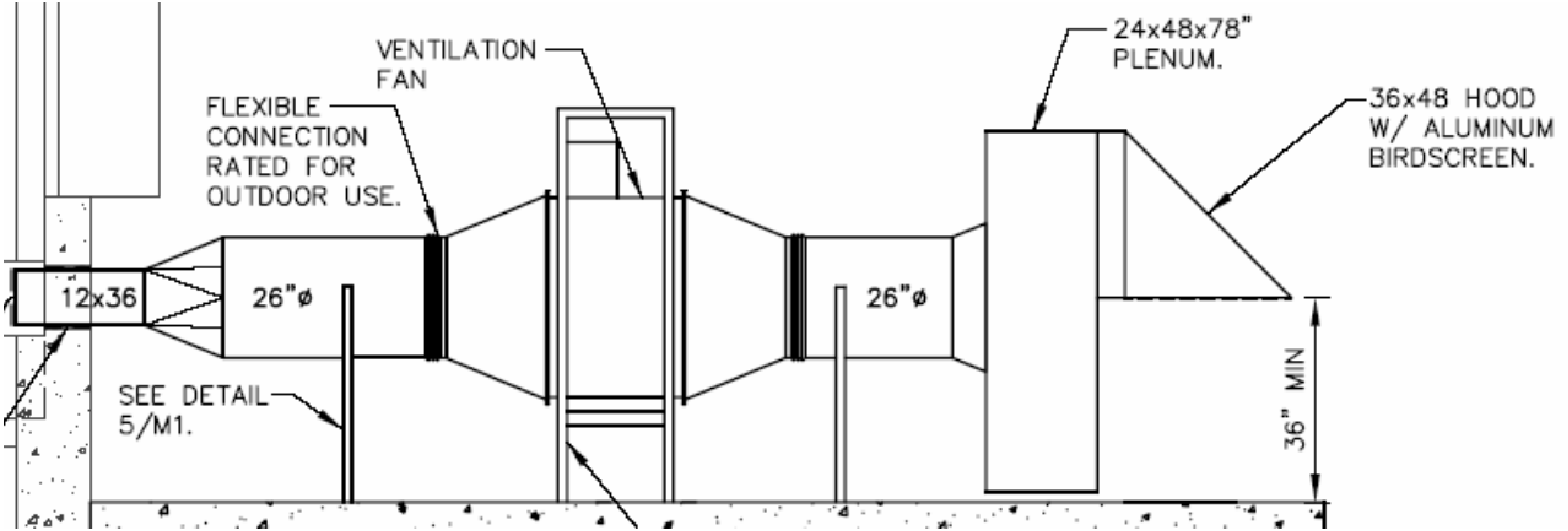
System Effect

- AMCA TECHSPECS VOL 1 – No. 1 Has more info
- Vol 11 – No1 has evase system effect factors

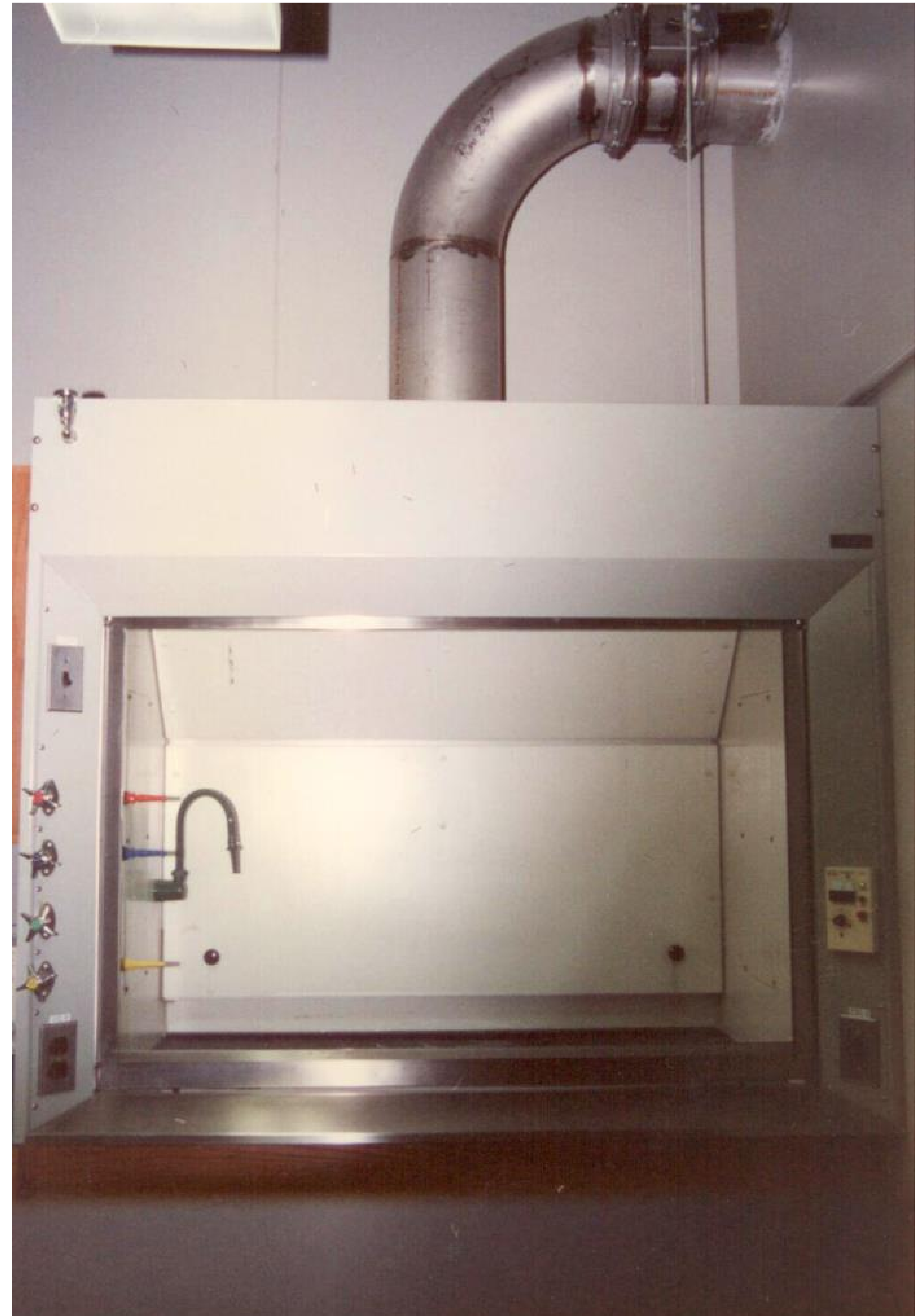


System Effect from Real Project

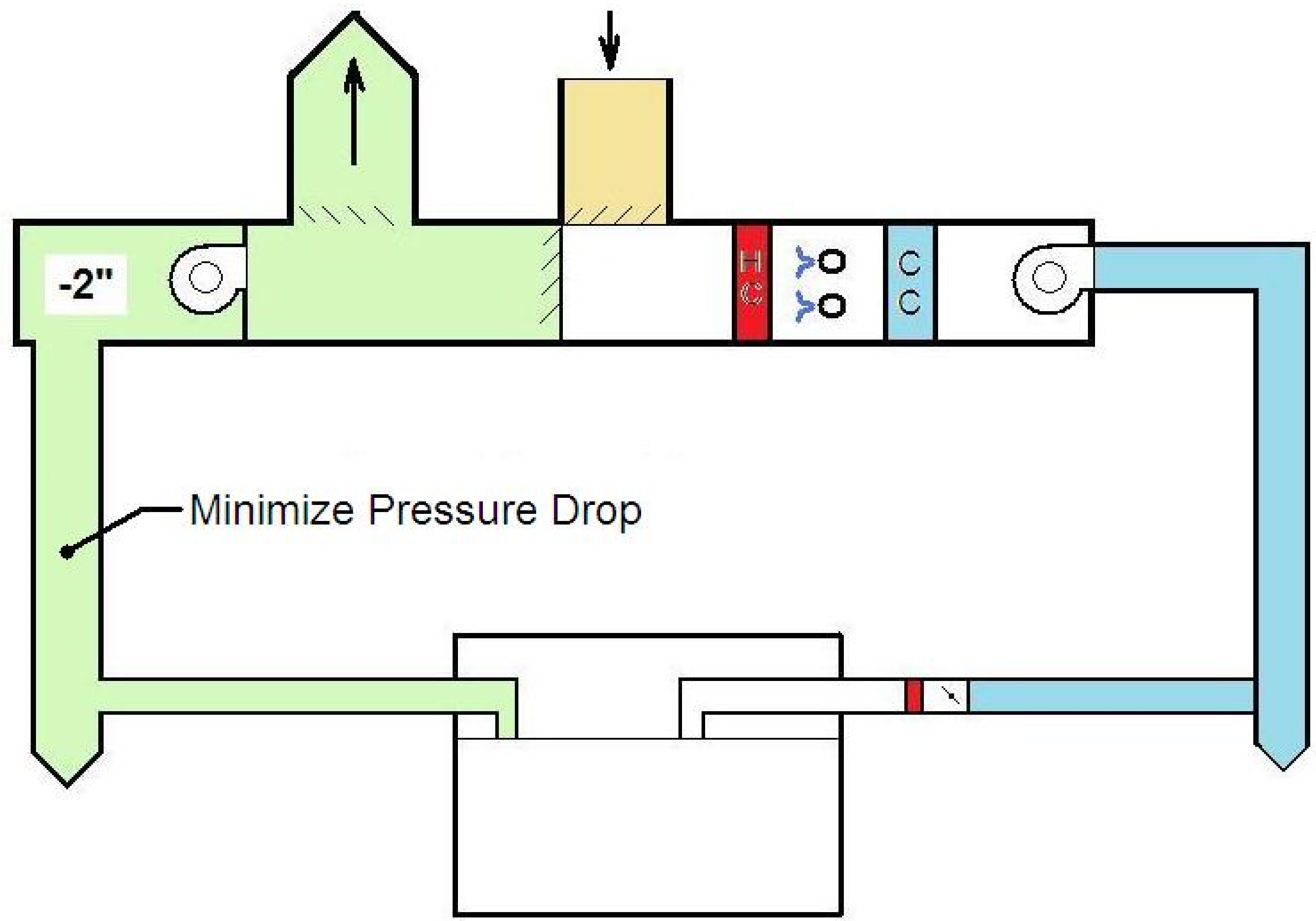
- Inline fan system effect

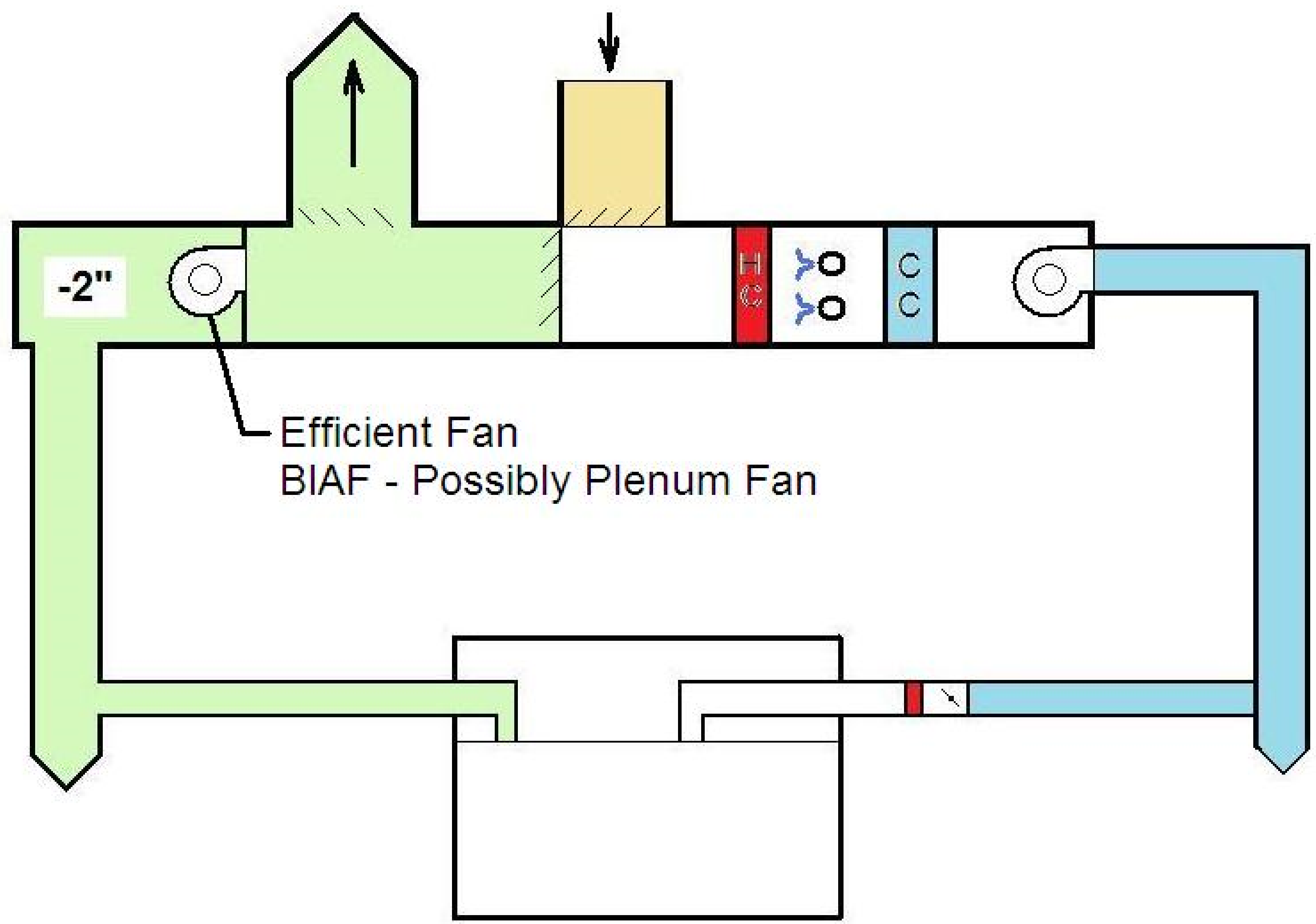


Original design for 5000 CFM at 0.5” wc, when balanced the fan produced approximately 2800 CFM at 0.2” wc.

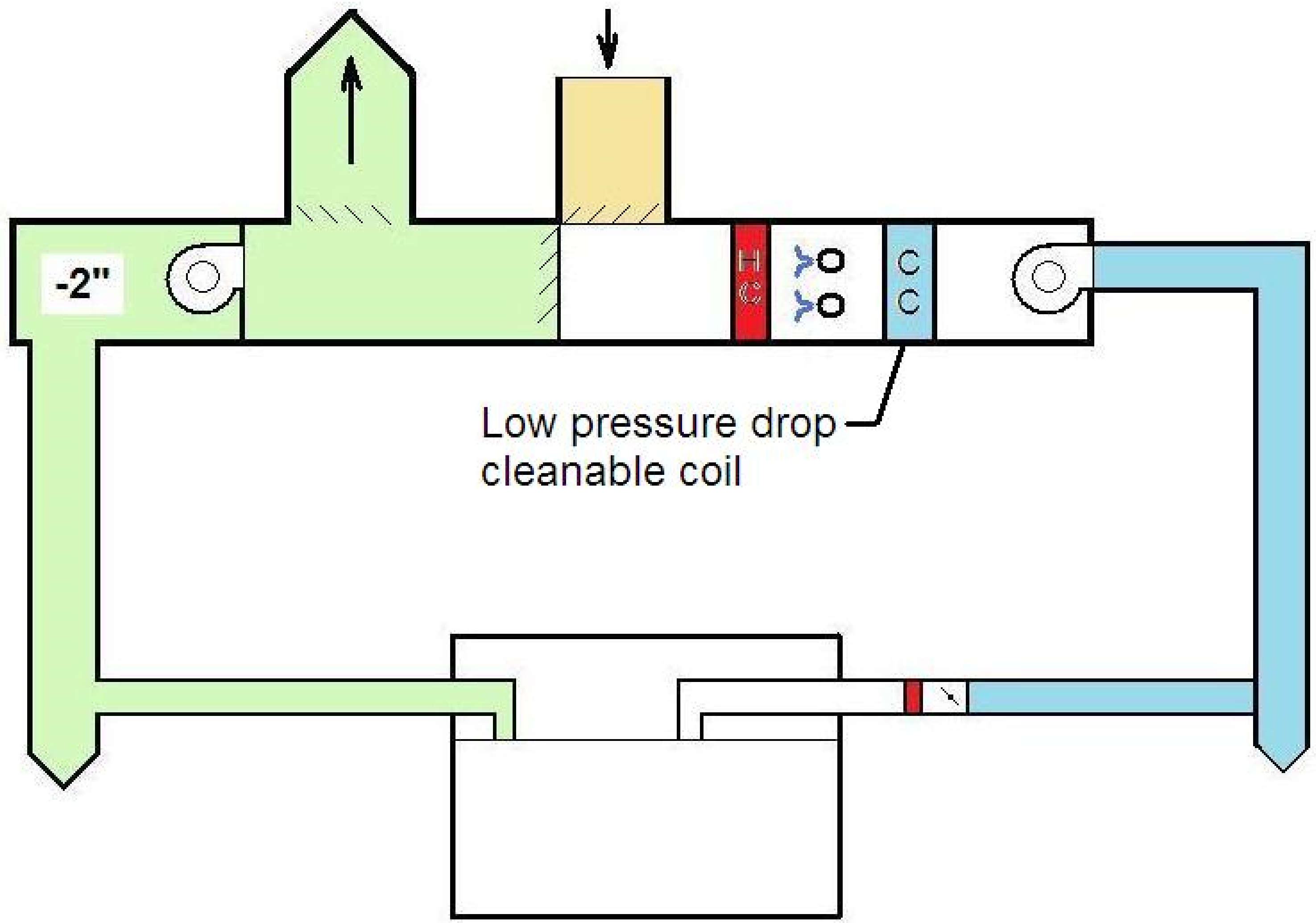



EX224
150
EX223
A 103
150

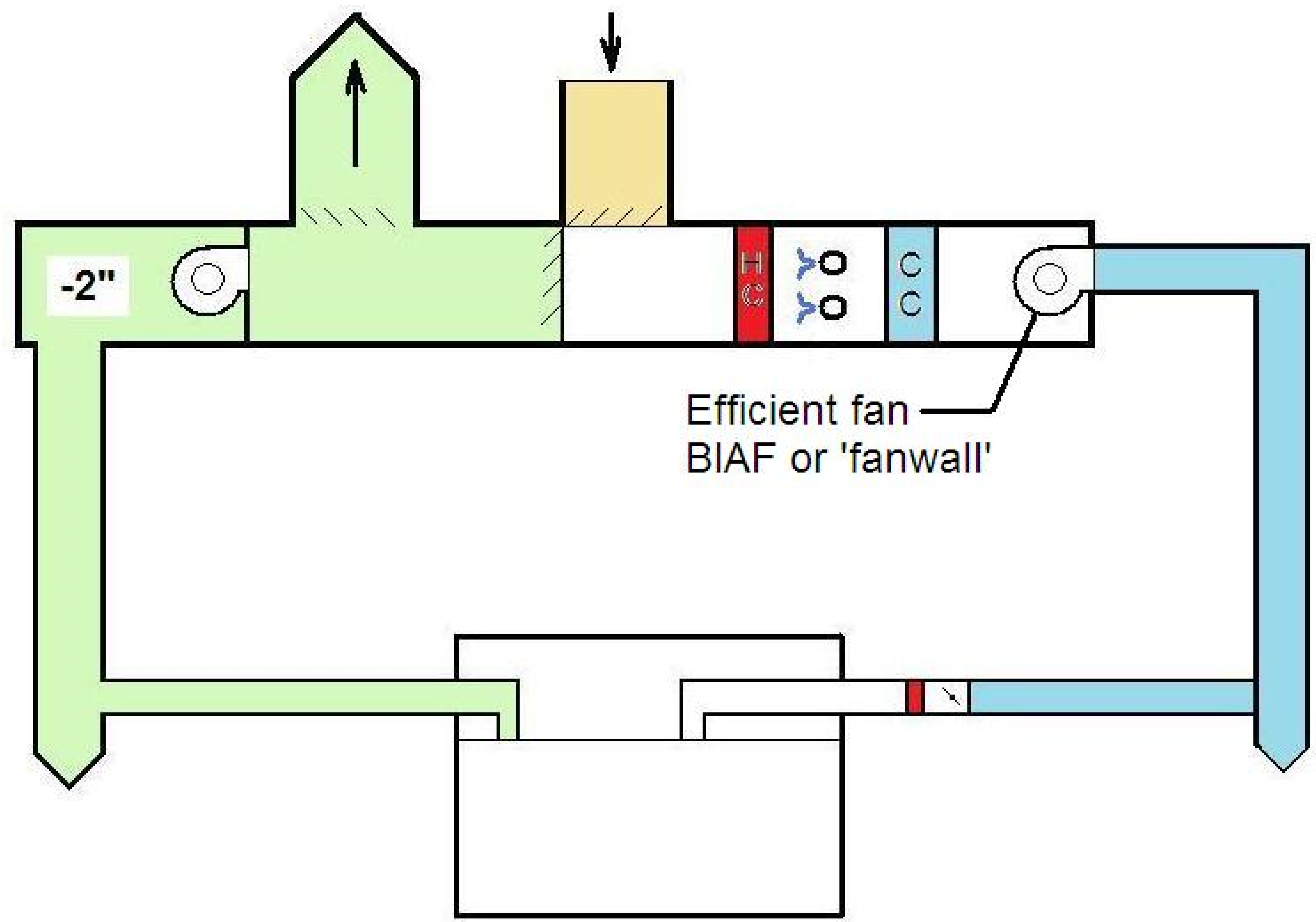

-2"
Minimize Pressure Drop



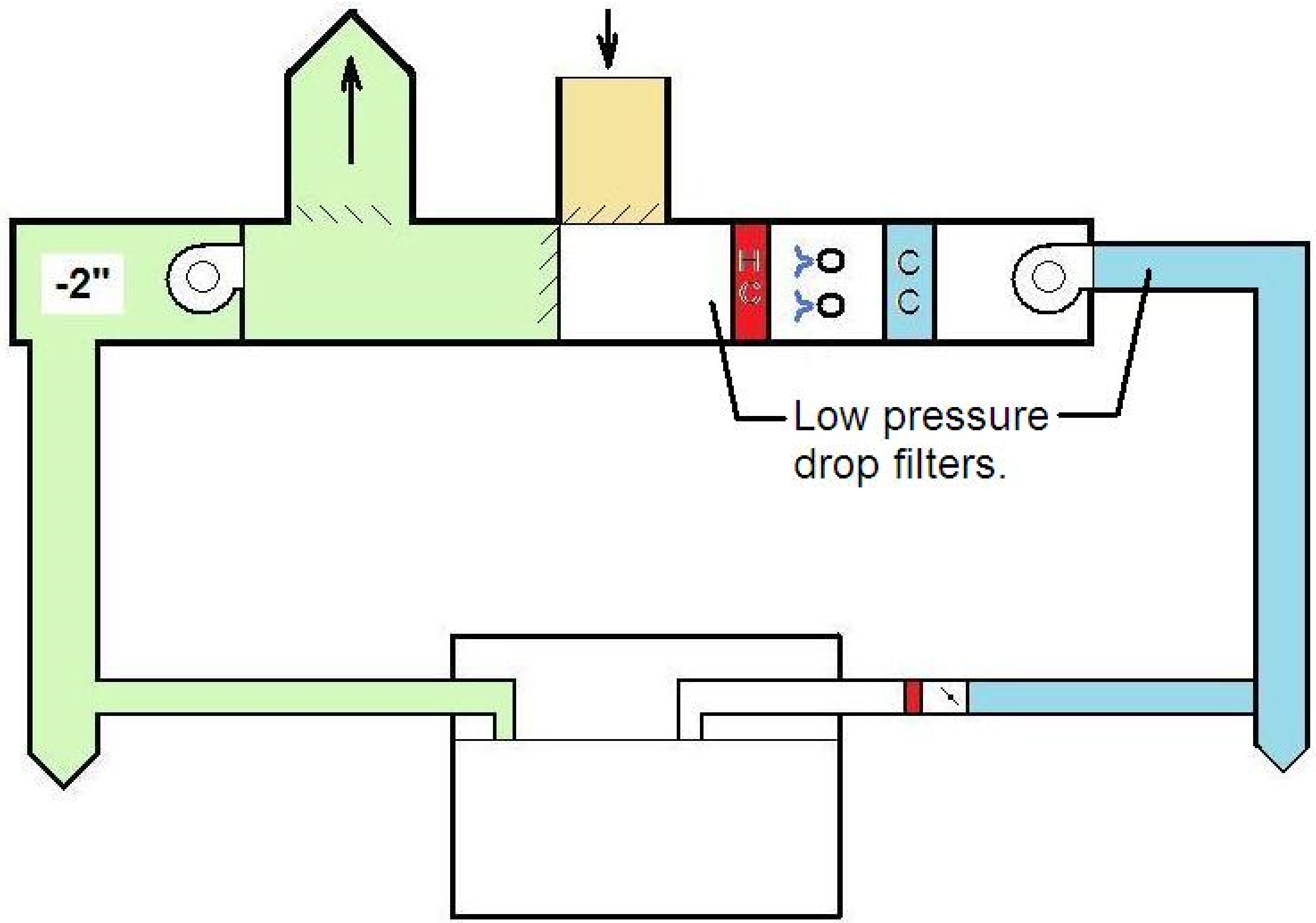
-2"
H
C
C
C
Efficient Fan
BIAF - Possibly Plenum Fan



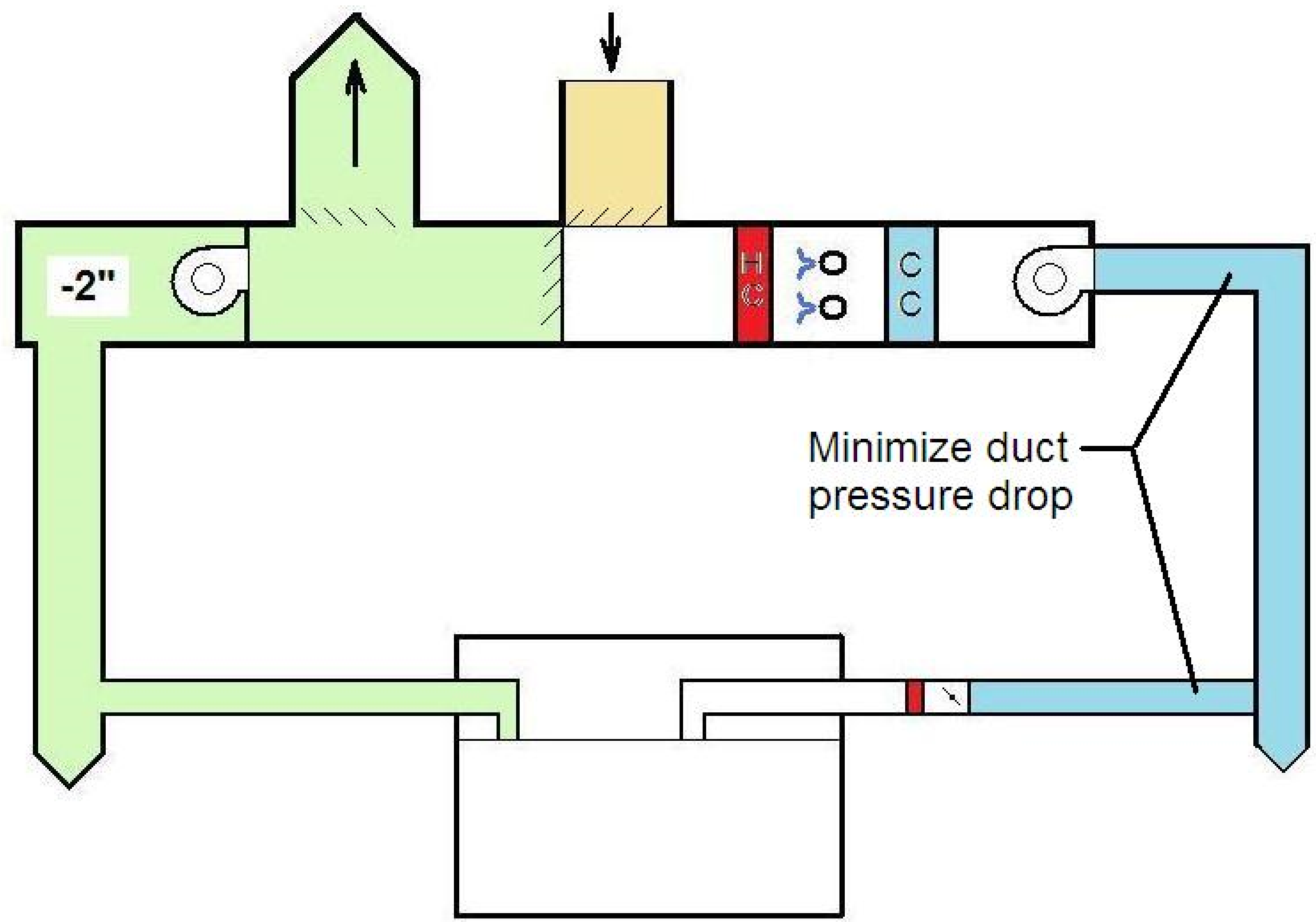
-2"
H
C
C
C
Low pressure drop
cleanable coil



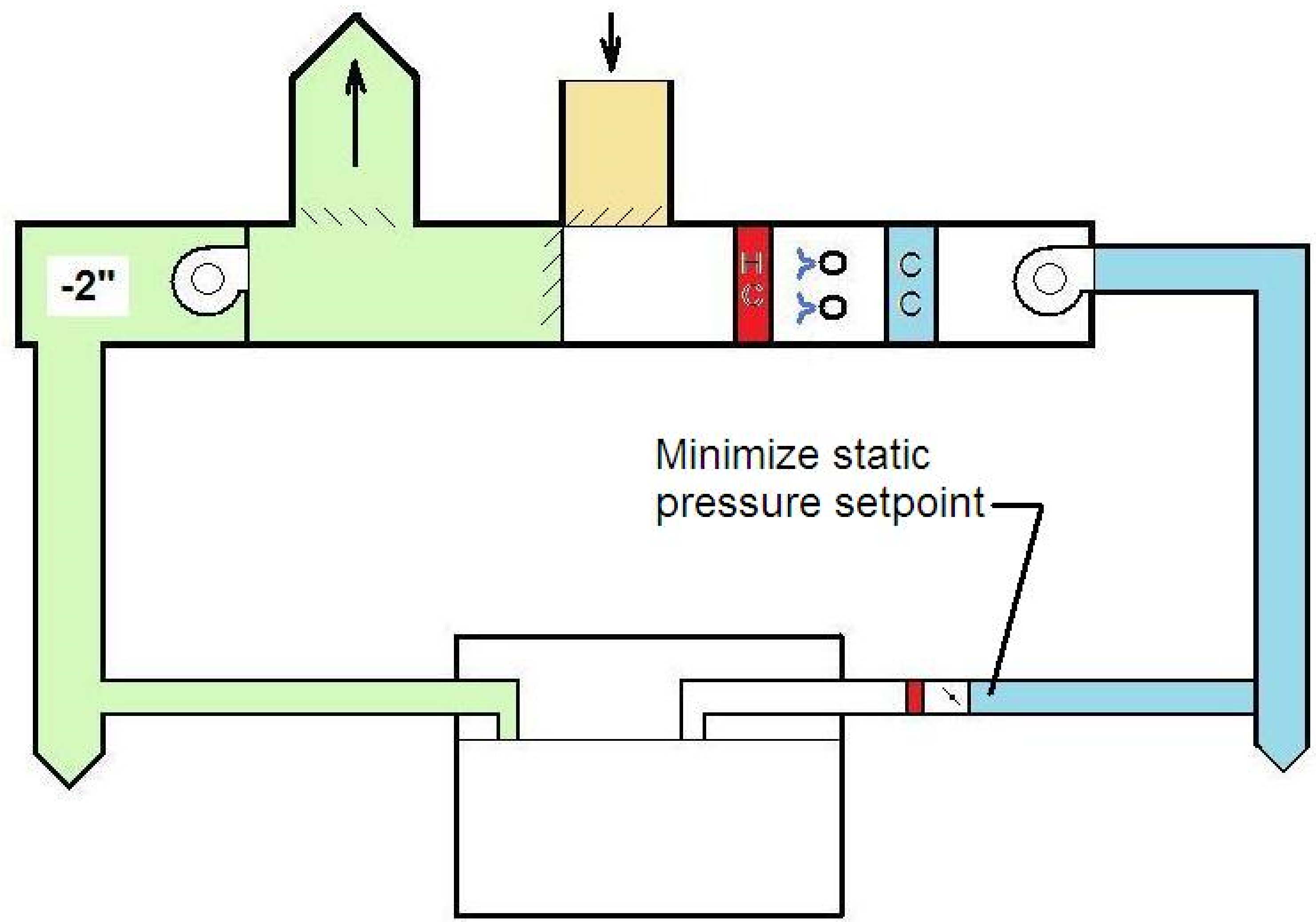
-2"
H
C
C
C
Efficient fan
BIAF or 'fanwall'



-2"
H
C
C
C
Low pressure
drop filters.

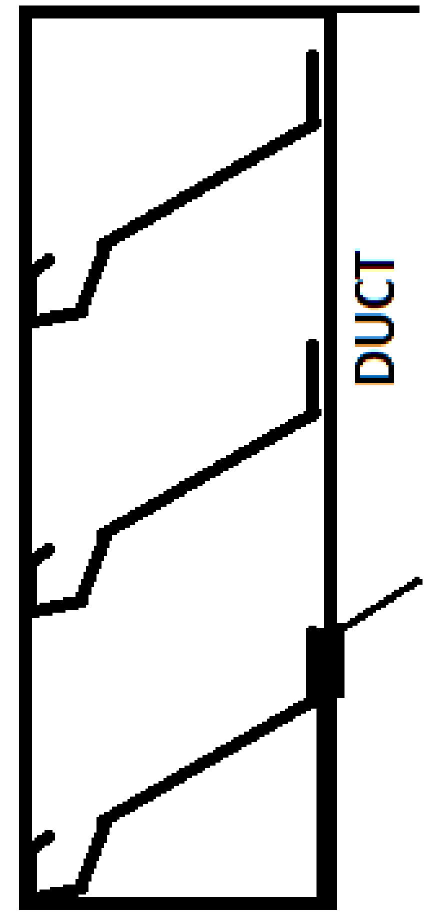


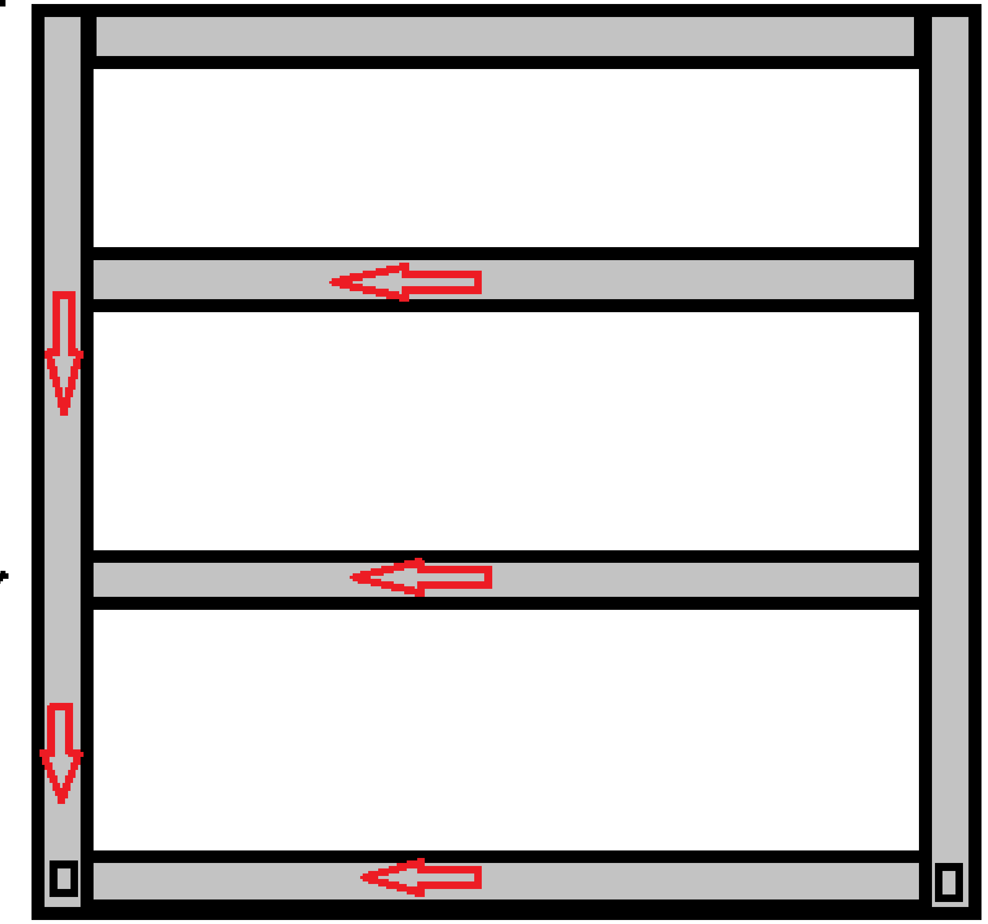
-2"
H
C
C
C
Minimize duct
pressure drop


-2"
H
C
C
C
Minimize static
pressure setpoint

Louvers & Gratings

- Drainable blade, also newer designs

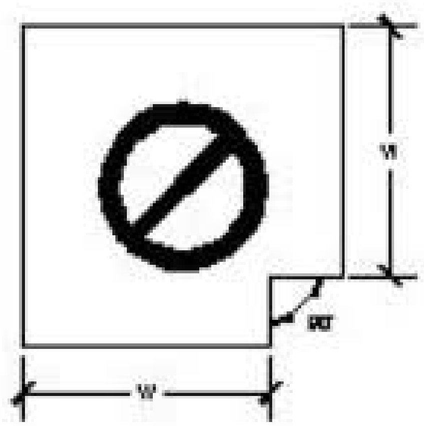




- Welded bar gratings

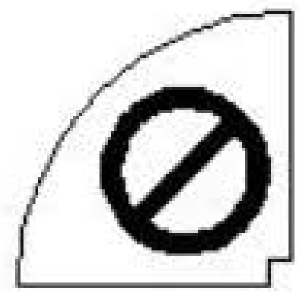
Free area Ratio	C-factor
0.30	6.2
0.40	3.0
0.50	1.7
0.55	1.3
0.60	0.97
0.65	0.75
0.70	0.58
0.75	0.44
0.80	0.32
0.90	0.14
1.00	0

Elbows



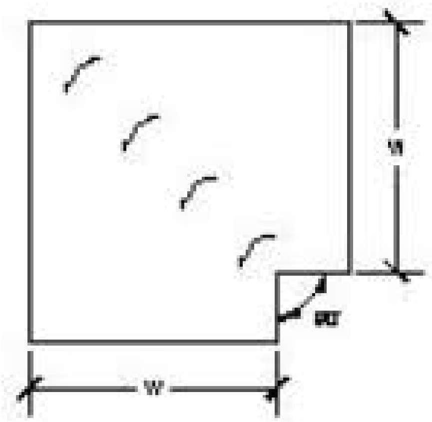
RECTANGULAR / OVAL / ROUND MITERED ELBOW WITHOUT VANES TYPE RE4

1.18



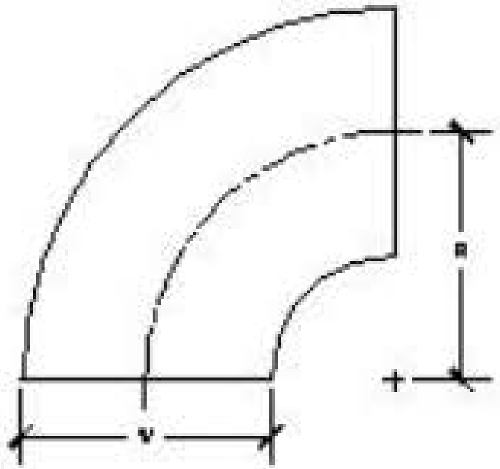
RECTANGULAR RADIUS ELBOW WITH SQUARE THROAT

1.18



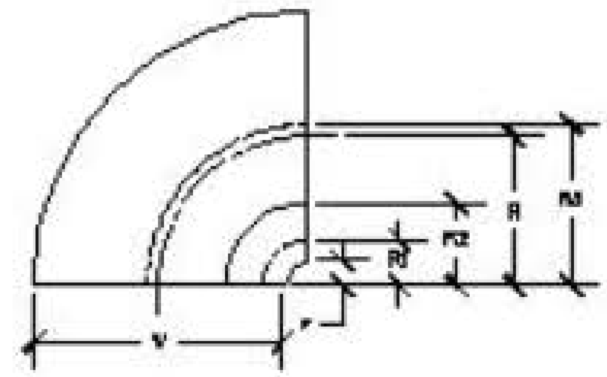
RECTANGULAR MITERED ELBOW WITH VANES TYPE RE2

0.11-0.41



RECTANGULAR RADIUS ELBOW TYPE RE1

0.20



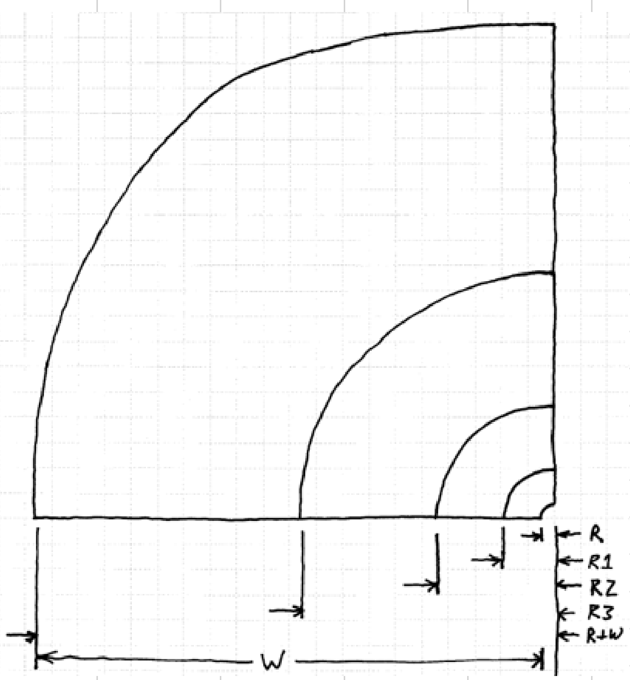
RECTANGULAR RADIUS ELBOW WITH VANES TYPE RE3

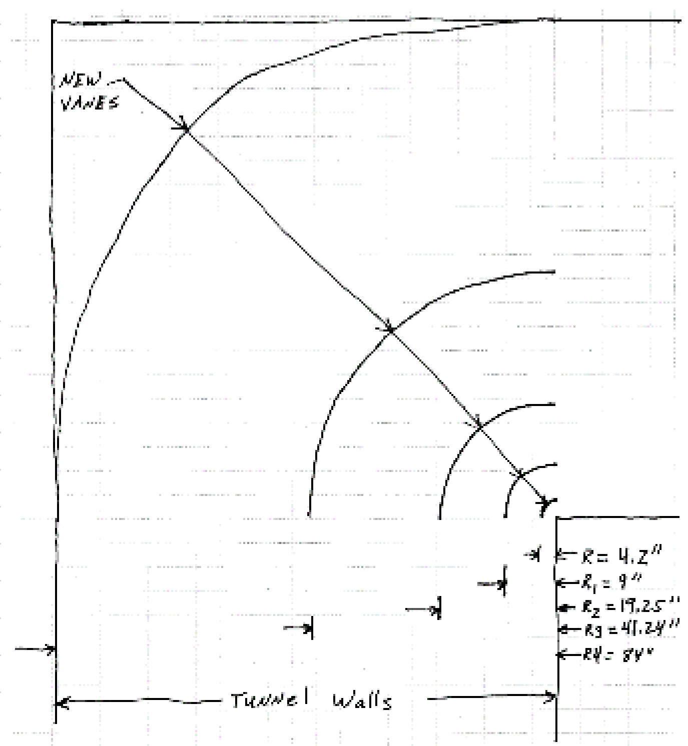
~0.10 Varies with r/W and Number of Vanes

Highest Pressure Drop → Lowest Pressure Drop

If the vanes will be added into an existing mitered duct or into a tunnel, use the second TAB

2	Number of Vanes
84	W - Duct Width (Perpendicular to Vanes)
36	H - Duct Height (Length of Vanes)
0.05	R/W
0.362	CR
88.20	Width of Duct + Inner Radius
4.20	R (Inner Radius)
11.59	R1 (First Vane
31.97	R2 (Second Vane)
-----	R3 (Third Vane &/or Outside Radius)
88.20	Outside Radius



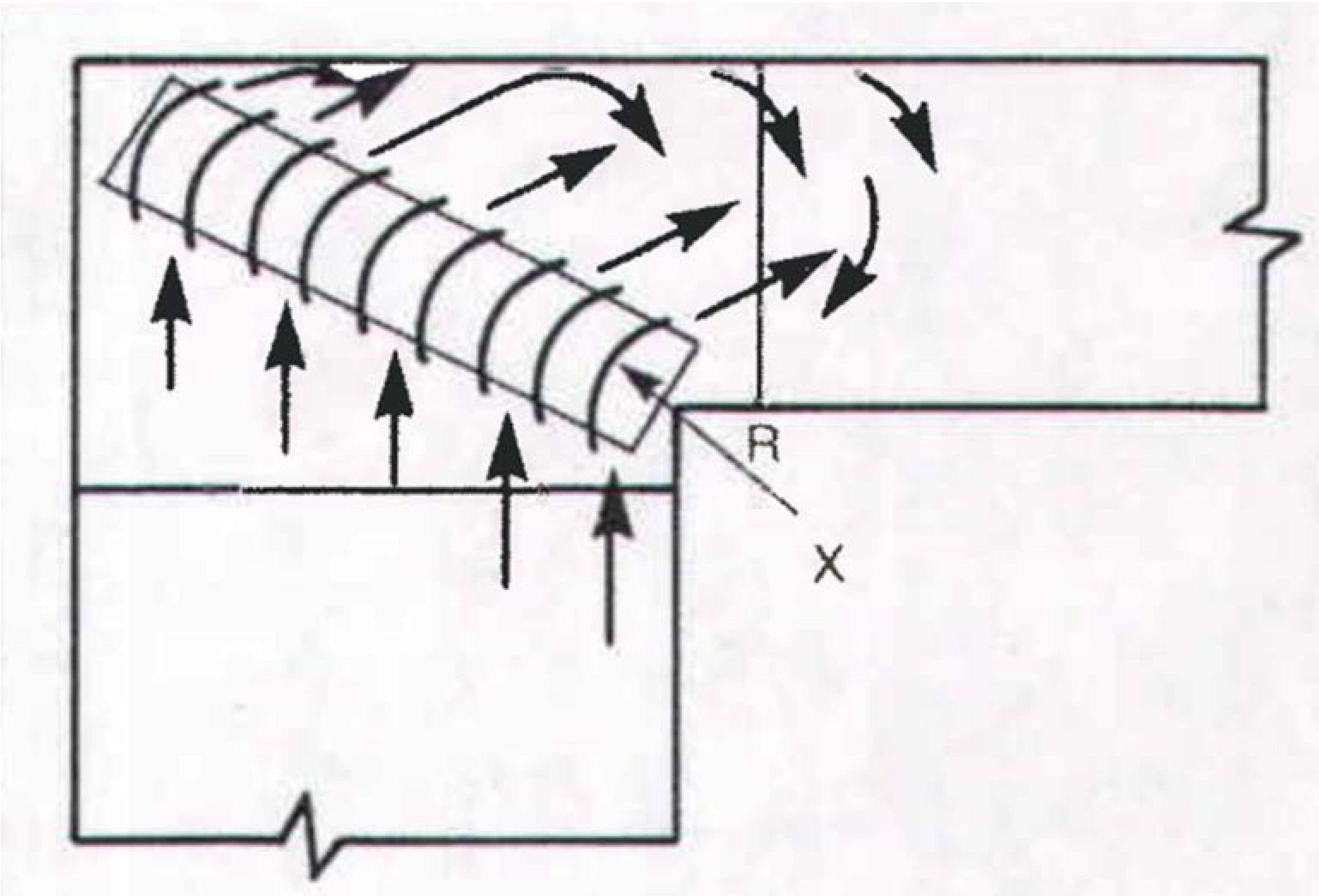


5.16.3 Vanes Missing

For many years contractors have eliminated every other turning vane from the vane runners installed in rectangular mitered duct elbows. Some contractors even believed that they would lower the pressure loss of the elbow by doing this. However, this practice more than doubles elbow pressure losses and is not recommended.

Figure 5-13 is a chart developed from SMACNA sponsored research tested single-thickness turning vanes. The distance between vanes was varied in increments of ¼ in. (6 mm). Airflow velocities varied from 1000 to 2,500 fpm (5 to 12.5 m/s). The pressure loss of the elbow with missing turning vanes was over 2.5 times the pressure loss of a properly fabricated elbow containing all of the vanes.

Proper Elbow Construction



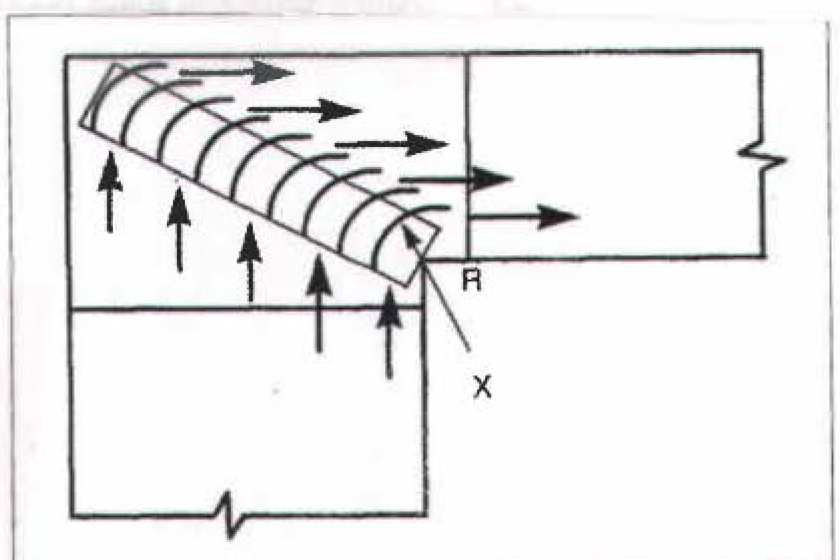
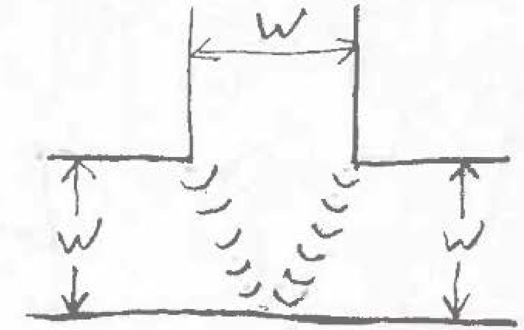
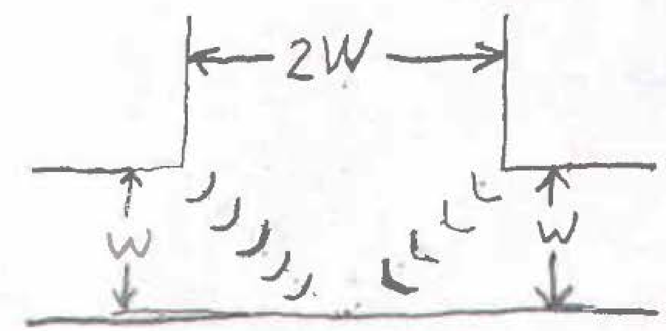


FIGURE 5-15 PROPER INSTALLATION OF TURNING VANES

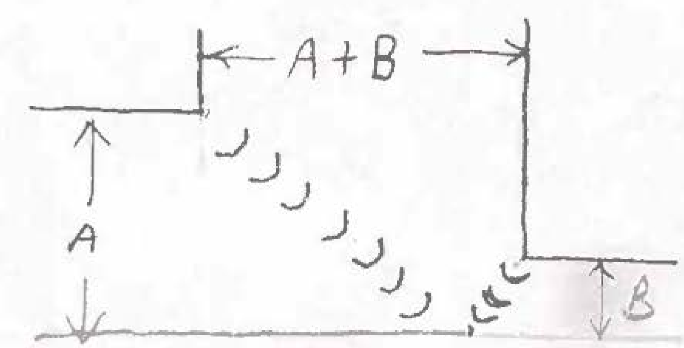
← ACCEPTABLE ONLY WITH SINGLE-THICKNESS VANES THAT ARE CUSTOM ADJUSTED TO ALIGN WITH THE FLOW



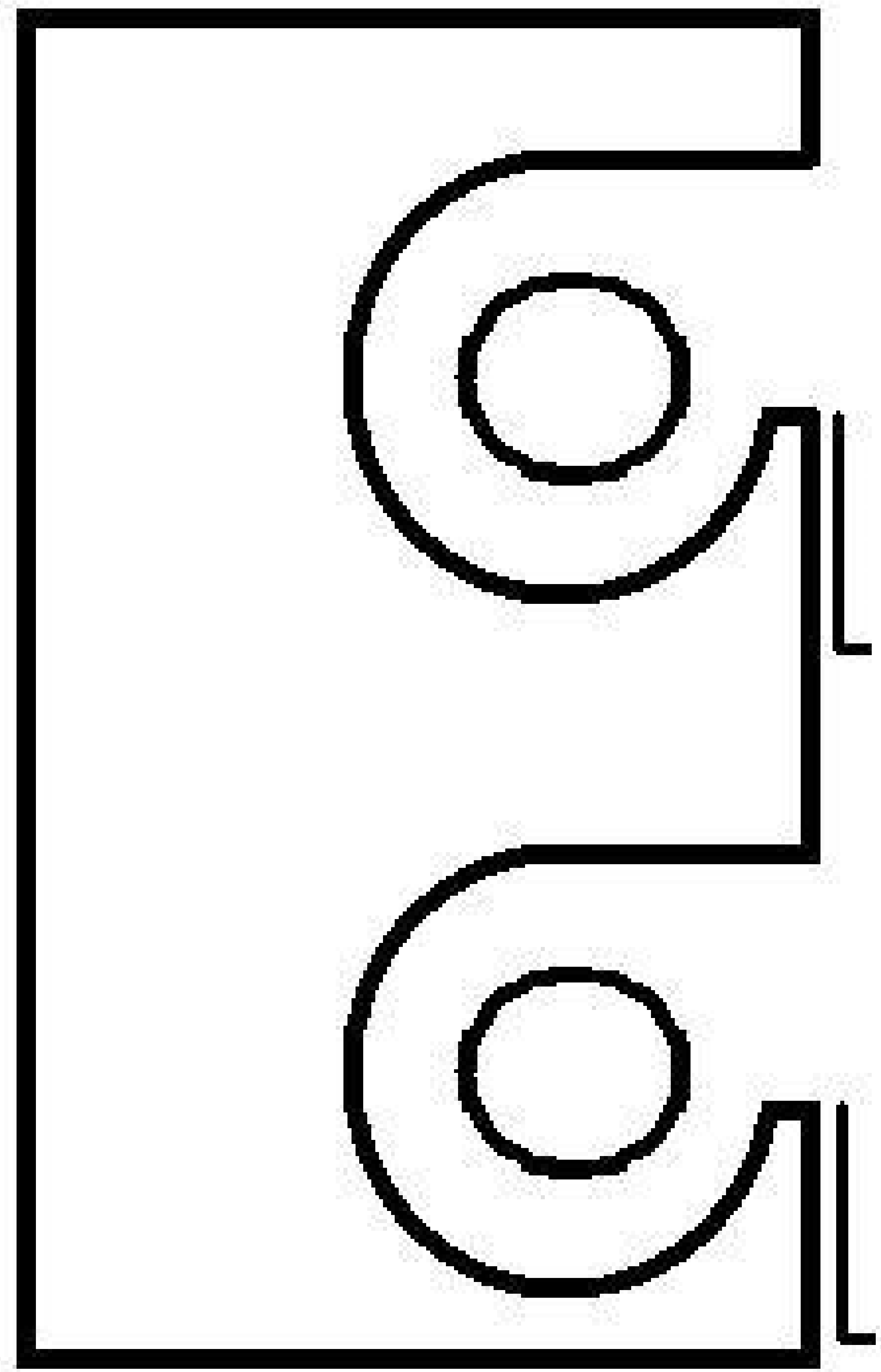
WRONG EXCEPT FOR SOME CUSTOM SINGLE-WALL VANE INSTALLATIONS

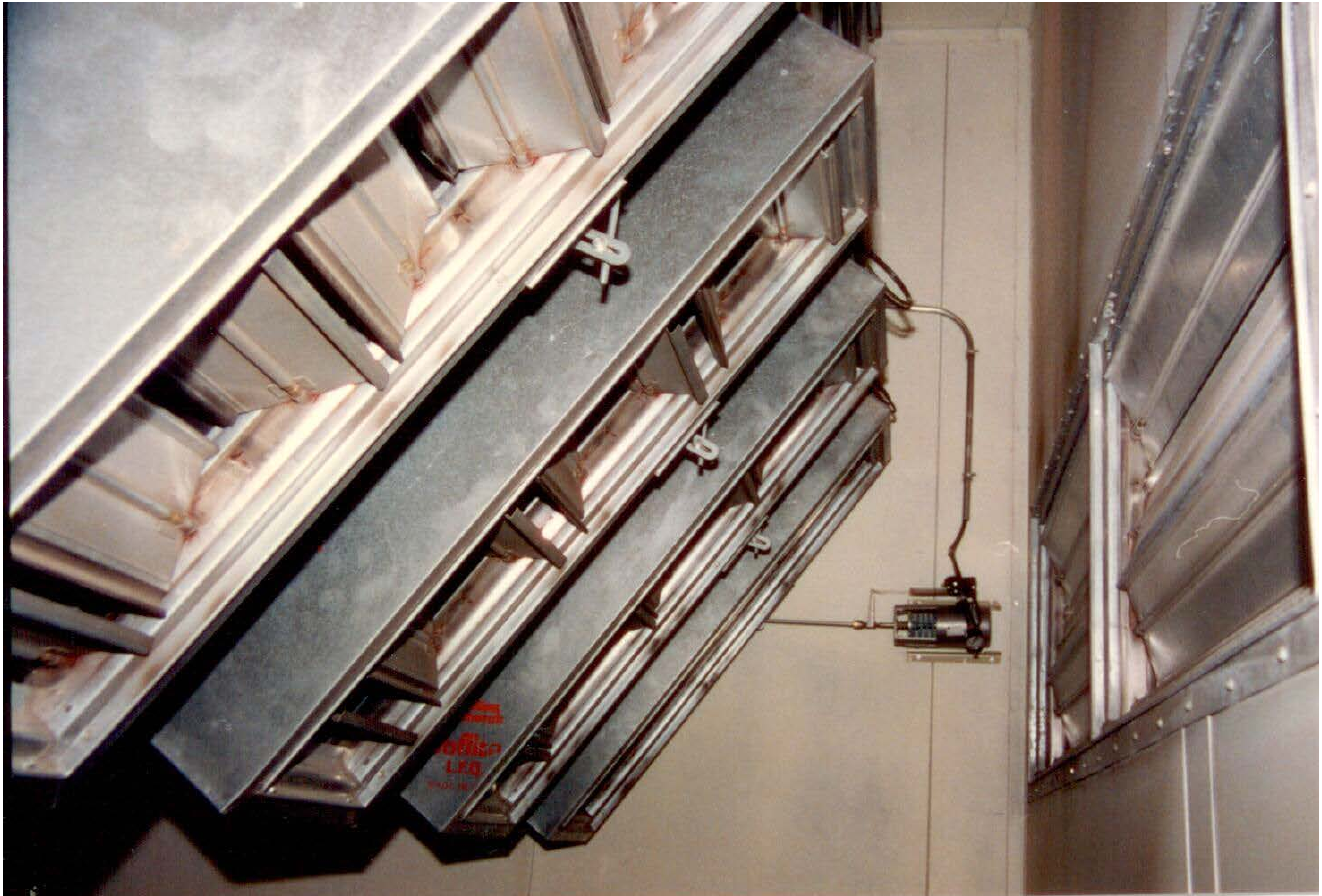


RIGHT

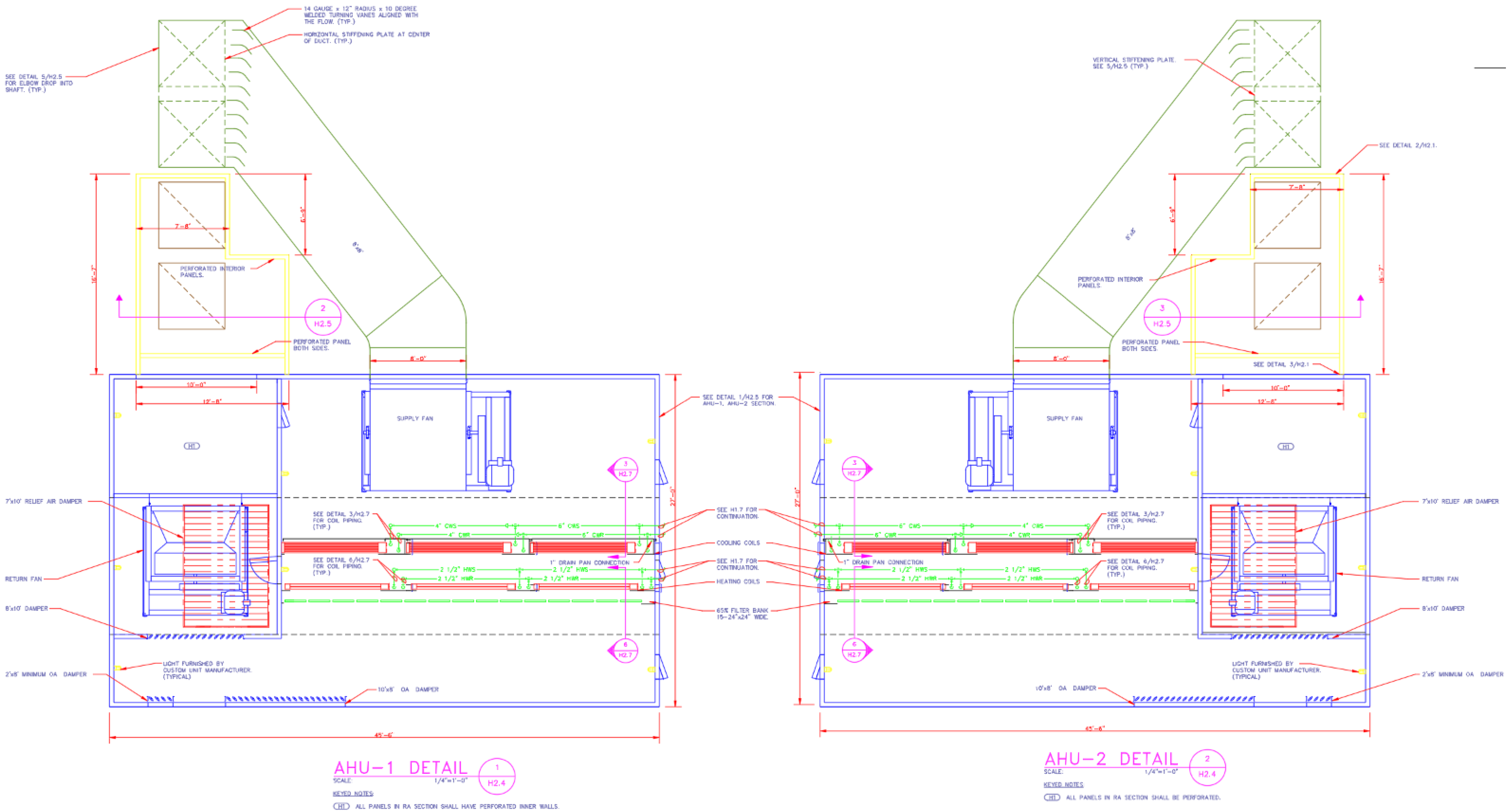


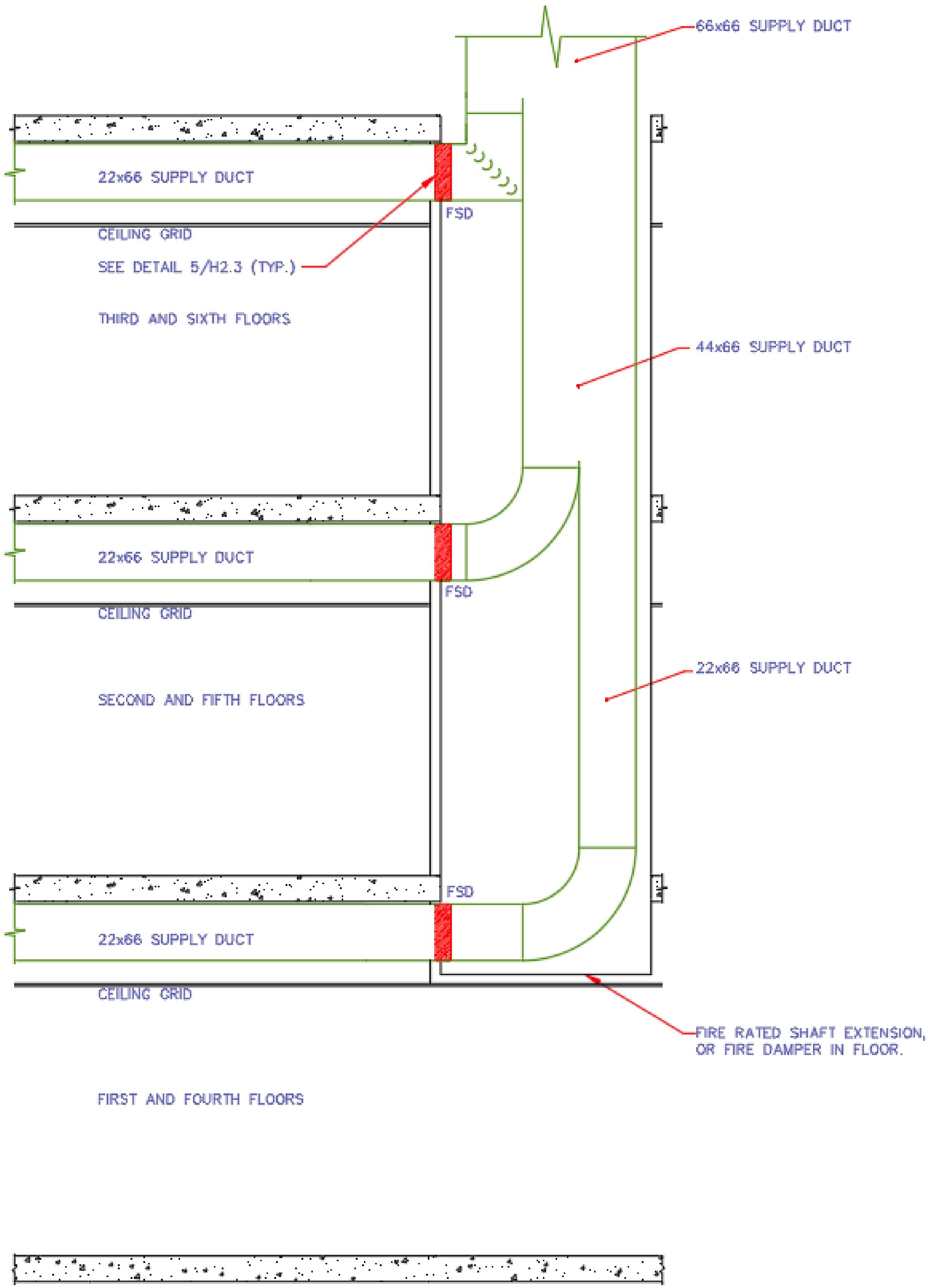
RIGHT





Think About Big Fittings

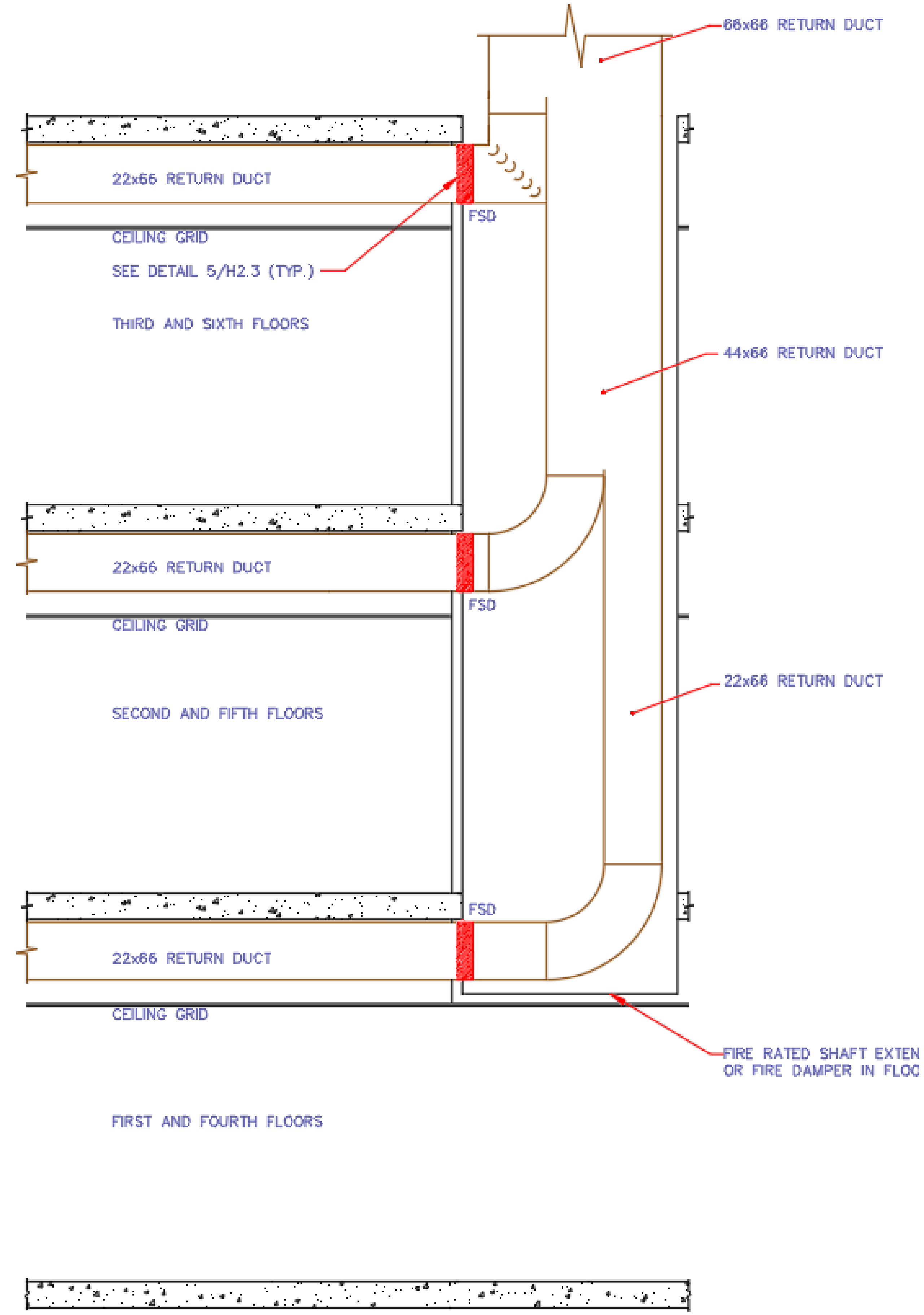




SUPPLY DUCT THROUGH CHASE

5 H2.1

SCALE: 1/4"=1'-0"

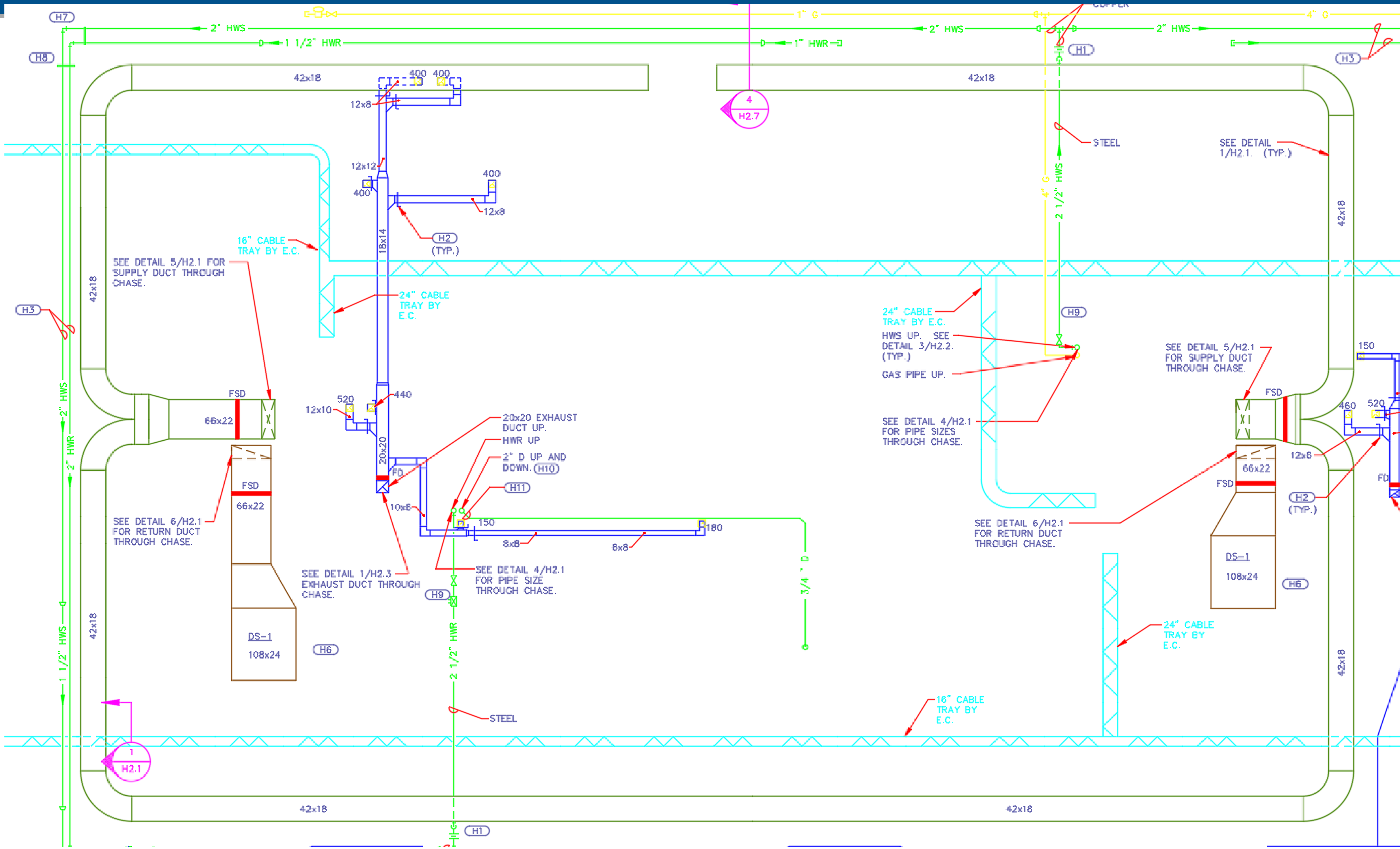


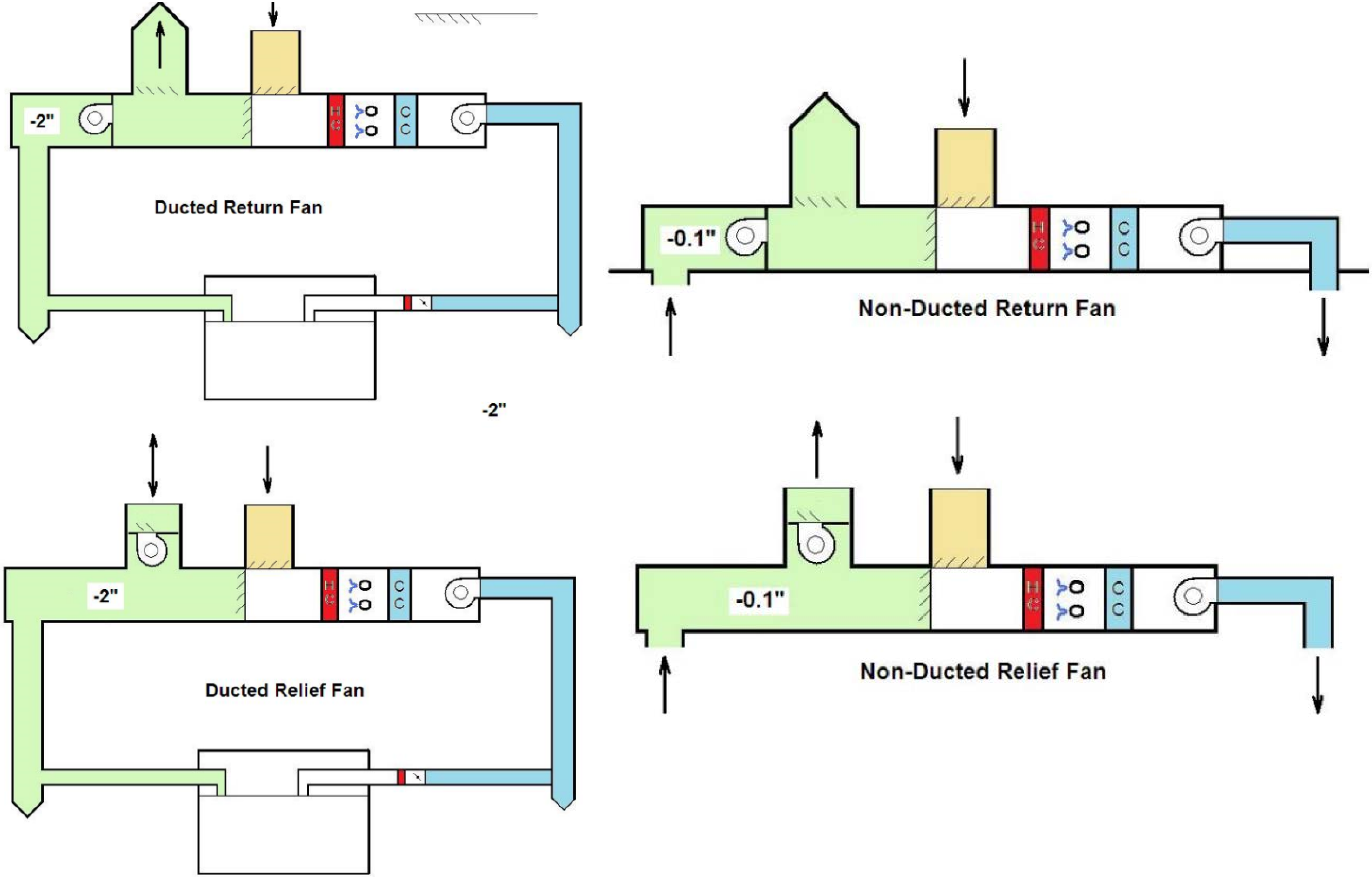
RETURN DUCT THROUGH CHASE

6 H2.1

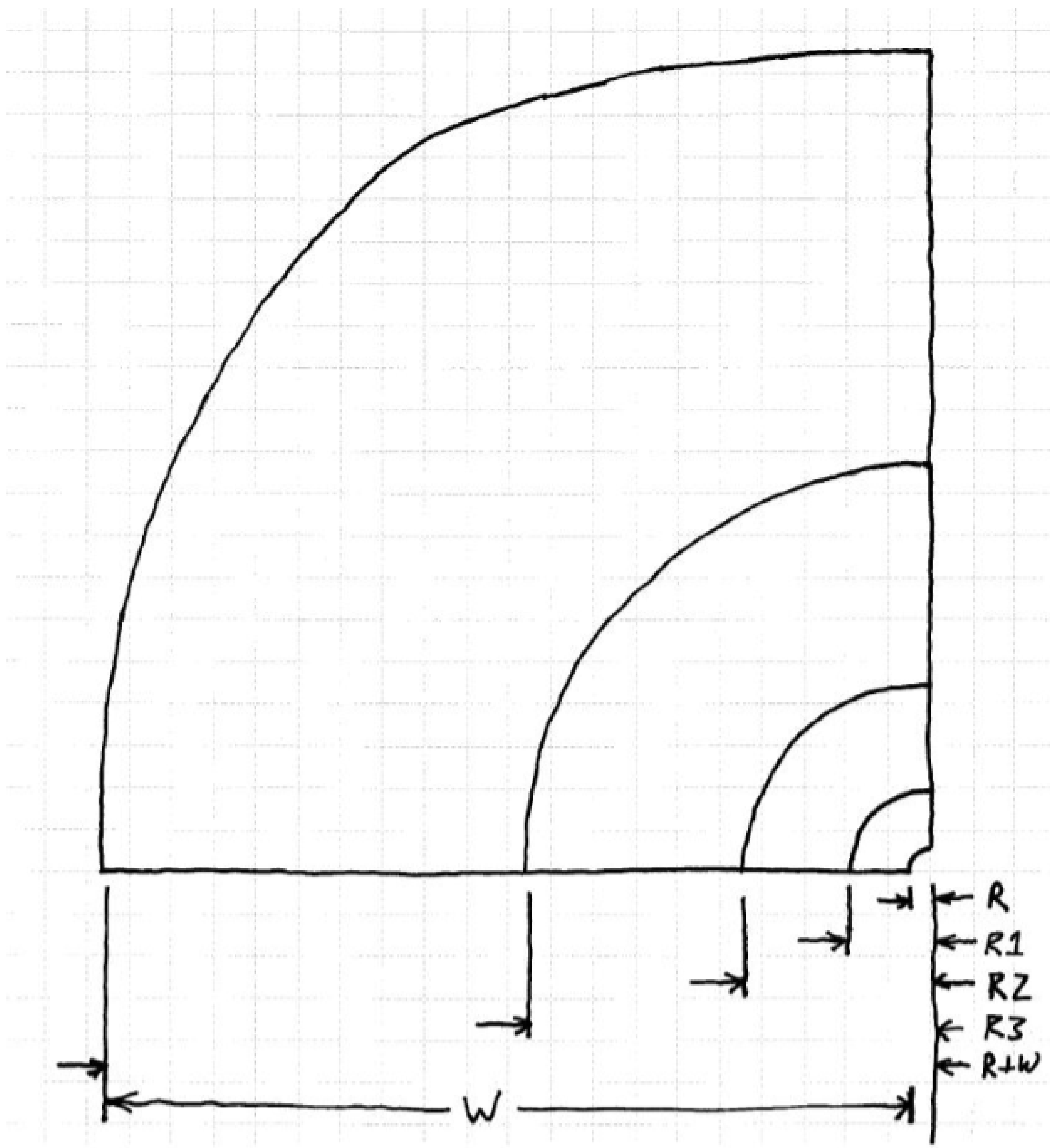
SCALE: 1/4"=1'-0"

A Few Good Fittings





-2"
Ducted Return Fan
-0.1"
Non-Ducted Return Fan
-2"
-2"
Ducted Relief Fan
-0.1"
Non-Ducted Relief Fan



R
R1
R2
R3
R+W
W


SERAC
Q7


DOH!


COOK

Questions / Discussion

Did I say that?!

Jeff.G.Boldt@IMEGcorp.com